JPL PUBLICATION 81-65

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Volume II: Status of Diesel Engine Technology

H. W. Schneider

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U.S. Department of Energy
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ABSTRACT

The engines of diesel cars marketed in the United States are examined. Prominent design features, performance characteristics, fuel economy and emissions data are compared. Specific problems, in particular those of NO_x and smoke emissions, are discussed along with the effects of increasing dieselization on diesel fuel price and availability, current R&D work and advanced diesel concepts.

The study generally concludes that diesel cars currently have a fuel economy advantage over gasoline engine powered cars. Many of the inherent diesel drawbacks (noise and odor) have been reduced to a less objectionable level. An equivalent gasoline engine driveability has been obtained with turbocharging. Diesel manufacturers see a growth in the diesel market for the next ten years. Uncertainties regarding future emission regulation may inhibit future diesel production investments.

With spark ignition engine technology advancing in the direction of high compression ratios, the fuel economy advantages of the diesel car is expected to diminish. To retain its fuel economy lead, the diesel's potential for further improvement must be used.

PREFACE

This report was prepared by the Jet Propulsion Laboratory for the U.S. Department of Energy, Office of Transportation Programs, for the Vehicle Systems Program managed by Albert Chesnes. This work was done at JPL in the Energy and Control Division by the Propulsion System Section as part of Vehicle Systems Tasks managed by Eugene Baughman.

The work was performed by the Jet Propulsion Laboratory and was sponsored by the U.S. Department of Energy under Interagency Agreements DE-AI01-76CS51011 and DE-AI01-80CS50194 through NASA Task RD 152, Amendment 165.

The purpose of this vehicle systems task was to perform a technical assessment of conventional automotive engine status and report the results. The status of the technology reported is that which was available through April 1981. This volume is part of the final report consisting of three volumes.

Volume I presents the status of Otto cycle engine technologies; Volume II presents the status of Diesel engine technology and Volume III compares these conventional engine types and discusses their future potential.

ABBREVIATIONS

A/F	air to fuel (mixture) ratio
BMEP	brake mean effective pressure
BMW	Bavarian Motor Works
BSFC	brake specific fuel consumption
CAFE	corporate average fuel economy
CARB	California Air Resources Board
CO	carbon monoxide
D	diesel model
DI	direct injection
DIN	Deutsche Industrie Normen (German Industry Standards)
EGR	exhaust gas recirculation
EPA	Environmental Protection Agency
GM	General Motors
g/mi	grams per mile
HC	hydrocarbon
hp	horsepower
IDI	indirect injection
Hz	Hertz, cycles per second
MB	Mercedes Benz
mpg	miles per gallon
MY	model year
NA	naturally aspirated
n/a	not available
NO_x	nitrous oxide
ODC	outer dead center
OPEC	oil producing and exporting countries
R&D	research and development
rpm	revolutions per minute
TC	turbocharge
TDC	top dead center
U.S.	United States of America
VW	Volkswagen

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CONTENTS

SECTION 1

INTRODUCTION AND SCOPE

The diesel engine is the most efficient mature heat engine available for current use. Based upon EPA ratings, the fuel economy of diesel powered cars ranges, on the average, from 25 to 40% higher than for emission controlled spark-ignition automobiles of about the same inertia weight and engine displacement. The breakthrough in high speed automotive diesel engines was brought about by Daimler Benz with the introduction of the divided chamber more than 40 years ago. Since then, Daimler Benz and a number of other well known companies, such as Peugeot, Perkins, Nissan, Ricardo, Fiat, and others, have been involved in developing and producing diesel engines for light trucks, taxi-cabs and other light duty vehicles where reliability and low operating cost are dominant factors.

In the past, the diesel engine's low power density, bulk and weight, slow response, cold start problems, and inconvenience caused by noise, smoke and odor, have deterred the private consumer from buying a diesel powered car. However, due to growing concern about the rising cost of gasoline and the possibility of acute shortages, the private interest in diesel cars has risen sharply since 1973. This and the corporate average fuel economy (CAFE) requirements, have spurred new developments that resulted in dramatic improvements in the small automotive diesel in regard to weight, size and driveability.

In addition to Daimler Benz and Peugeot, who supplied most of the diesel cars sold in the United States, a newcomer in the diesel field, Volkswagen (VW), began offering a diesel option for their subcompact Rabbit line in 1977. General Motors (GM) became the first U.S. producer to offer a gasolinengine-based V-8 diesel as an option in their 1979 model pickup trucks, Olds Cutlass and Cadillac Eldorado cars. Other large producers such as Ford, Chrysler, and Bavarian Motor Works (BMW) are planning to have diesel options available in the near future. It is expected that by 1985, 25% or more of all cars sold in the United States will be diesel powered.

Unfortunately, the sales outlook for diesel cars is overshadowed by factors that may inhibit their widespread use in the United States and other free-world countries. Present diesel automobiles can meet current emission standards, but their ability to meet future, more stringent emission requirements regarding particulates and nitrous oxide (NO_x) is still an unresolved problem. Concern is mounting that particulates and organic matter emitted from diesel engines may be carcinogenic and hazardous to health. Changing refinery production to produce more diesel fuel results in higher costs that would have to be passed on to the diesel consumer unless regulated otherwise. There are also limits as to how much diesel fuel can be produced from each barrel of crude oil.

A variety of private- and government-funded diesel oriented research programs are under way, primarily in Europe, the United States and Japan. Current efforts concentrate mainly on improving and on learning more about the physics of diesel combustion and emissions, the causes of smoke and

odor, as well as the effects of engine operational and design parameters on emission characteristics. Approaches are also being studied to entrap undesirable particulate pollutants that cannot be controlled by other means.

Due to technology improvements made during the last 20 years in the small turbine and compressor field, turbocharging (TC) has become practical for use in small automotive diesels. Turbocharging tends to reduce NO_x, smoke and odor, and substantially raises the power-density (power/displacement) and driveability of diesels up to or beyond that of emission controlled spark-ignition engines. Turbocharged diesel engines are currently marketed by Daimler Benz, Peugeot, and Audi. BMW will be the only producer to offer a TC diesel without the option of a naturally aspirated version.

Beyond TC, the high speed diesel engine still has large potential for further improvement. Current thinking in advanced engine research is aimed at using more refined injection techniques (programmed injection), direct injection (DI) open chamber combustion, variable compression ratio, ceramic lining of combustion chamber walls to reduce cooling losses, and the possible use of ceramic cylinders and pistons, a technology that is currently being pioneered by Cummins. The reduction of cooling losses results in higher internal wall temperatures that tend to reduce the formation of particulates, and make the application of turbocompounding techniques more effective and perhaps economically feasible for cars.

The use of exhaust heat by means of an organic Rankine engine (bottoming cycle) is being developed for use on trucks and large diesel engines. Because it requires relatively complex mechanical and control systems, bottoming does not seem to be an economically feasible approach for light and medium duty vehicles at this time. However, some privately sponsored efforts in this direction are being made.

SECTION 2

GENERAL CONSIDERATIONS

2.1 TYPICAL ENGINE DESIGN FEATURES AND OPERATING CHARACTERISTICS

Both the Otto and the diesel engine derive useful work from combustion under pressure, followed by expansion and force acting on a piston. The fundamental difference between these two engine concepts lies in the way the fuel is introduced, and how combustion is brought about and controlled.

Otto engines ingest a mixture of a combustible fuel vapor and air, in which the latter serves as the oxidizer as well as a working fluid. Combustion and heat release are initiated near the end of the compression phase with an electrically ignited fuel charge. Therefore, the Otto engine is frequently referred to as the "spark ignition engine" or, because it is commonly operating on volatile petroleum fuel, as the "gasoline engine." After ignition, the combustion process takes its course and cannot be further controlled. The heat release is usually very rapid and takes place at almost constant volume conditions.

The diesel engine ingests only air. Fuel is introduced as an atomized spray and mixed with the air by forced injection near the end of the compression phase. Ignition of the combustible mixture is induced completely by the temperature rise resulting from the compression of air. Therefore, the diesel engine is frequently referred to as the "compression ignition" engine.

One of the reasons that diesel engines are more fuel efficient than spark-ignition engines is that higher compression ratios can be obtained without knocking. The theoretical fuel efficiency of both the Otto and the diesel engine improves with increasing compression ratio. However, weight and friction penalties associated with high pressure and structural loads, plus cold starting requirements, are the primary factors that determine the practical compression ratio of a diesel engine. The best efficiency for diesels is obtained at compression ratios on the order of 16 to 18. In present automotive diesels, the compression ratio required for reliable starting is about 22 to 1.

The strongest contributor to the diesel's superior partload fuel economy is the absence of an inlet throttle. Instead of throttling the inlet flow, diesels are controlled by metering the fuel input per working cycle and cylinder according to the load level desired. Diesels burn about 1.2 to 4 times leaner than Otto engines and generally operate at lower combustion temperatures. Cooling losses are reduced and less exhaust energy is wasted in comparison to the Otto engine.

However, the relatively lean combustion of the diesel engine is also the primary cause for one of the major disavantages: a reduced power output per unit displacement. Also, the higher compression ratio of diesel engines requires a stronger and heavier structure in order to withstand the higher

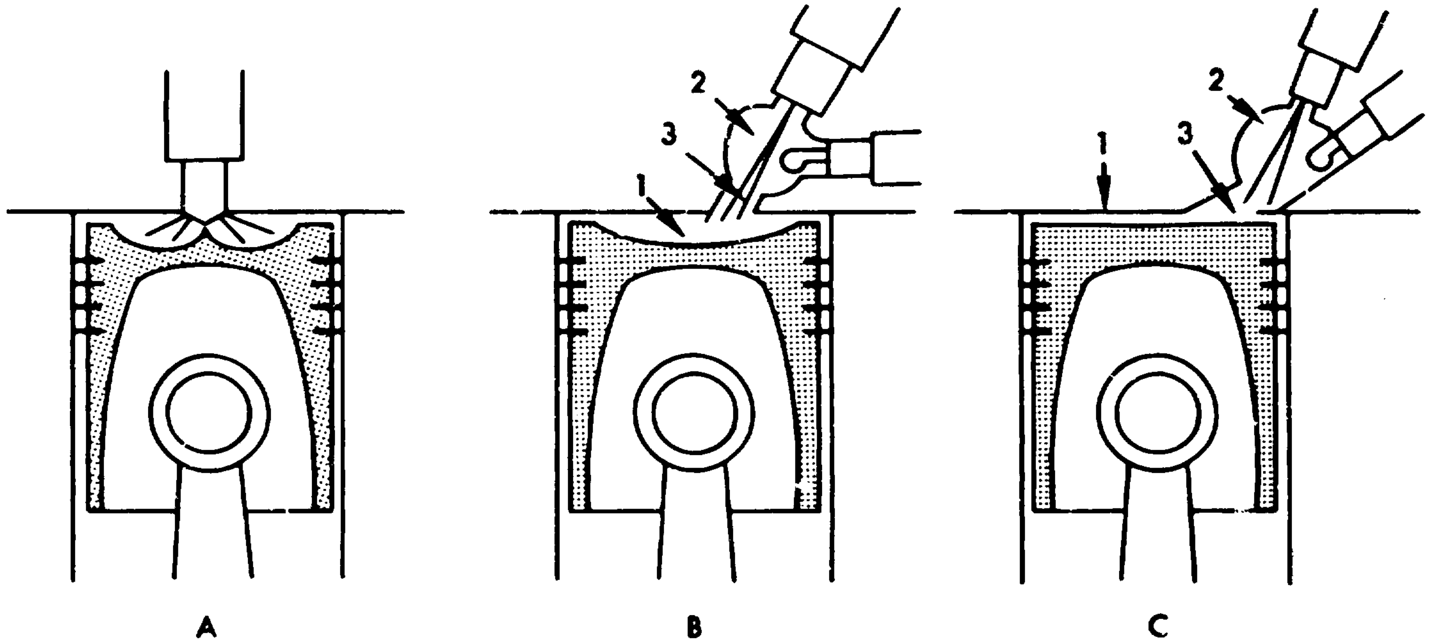
internal stresses. A stronger crankshaft, and in engines with less than six cylinders, more flywheel mass is needed which impairs the inertial response of the engine.

Most of the large commercial diesels built today are open chamber, direct injection engines (Fig. 2.1-1A), requiring high injection pressures in order to obtain sufficient penetration while simultaneously keeping the droplet size small. A variety of chamber designs has emerged over the years in an effort to improve combustion efficiency and to moderate the initial sudden heat release and pressure rise, which is the primary cause for the diesel's harsh dynamic behavior and noise generation. Nozzle orifice size, injection pressure, fuel line and associated "waterhammer" effects are the primary factors that limit minimum cylinder displacement and maximum engine speed. Although new and promising approaches to the open chamber "knock" problem are underway, the open chamber direct injection concept is currently only in use on larger engines with shaft speeds below 2500 rpm; and with a minimum cylinder displacement on the order of 1.

The breakthrough for smaller cylinder displacements and higher shaft speeds was brought about by Daimler Benz about 40 years ago with the introduction of the divided chamber. As shown in Figure 2.1-1B, in the divided chamber engine, a part of the compressed air is displaced from the main chamber (1) through the throat (3) into the pre-chamber (2). Fuel is injected only into the pre-chamber (2) where most of the combustion occurs. The pressure rise associated with the heat release forces the hot gases through the throat (3) and back into the main chamber where they expand to deliver useful work to the piston while further combustion takes place.

The pre-chamber concept has advantages and disadvantages. It allows the small high-speed automotive diesel engine to be practical at the expense of engine efficiency, while achieving reduced structural weight, reduced noise, improved driveability and greater convenience. The main advantage provided by the pre-chamber is that combustion and pressure spikes are confined to a small space, with a more favorable surface-to-volume ratio than is feasible in open chamber engines. As can be seen from Figures 2.1-2, 2.1-3, and 2.1-4, fuel penetration distance, injection pressure, heat losses, and ignition delay times are therefore greatly reduced. Usually, only one or two nozzle orifices are required to produce an acceptable distribution and atomization of the fuel. The pressure spikes associated with the initial heat release are partially absorbed by the more spherically shaped structures forming the pre-chamber before they reach the main chamber and the piston. Consequently the pressures acting on the large cylinder and on the power train are reduced, and stresses and noise level are considerably lower than those produced by open chamber engines.

A further step toward reducing ignition delay and associated pressure spikes, lowering injection pressures and raising the speed capability, was obtained by off-setting the throat axis relative to the prechamber so that a swirl is produced during the compression phase, as shown in Figure 2.1-1C. The swirling motion of the incoming air forces the residual gases toward the center of the chamber, and feeds fresh air past the fuel nozzle in a basically controlled and predictable manner. This further reduces the neces-



A - DIRECT INJECTION CHAMBER
B - PRE-CHAMBER
C- SWIRL CHAMBER

1 - MAIN CHAMBER
2 - PRE-CHAMBER
3 - THROAT

Figure 2.1-1. Typical Diesel Combustion Systems

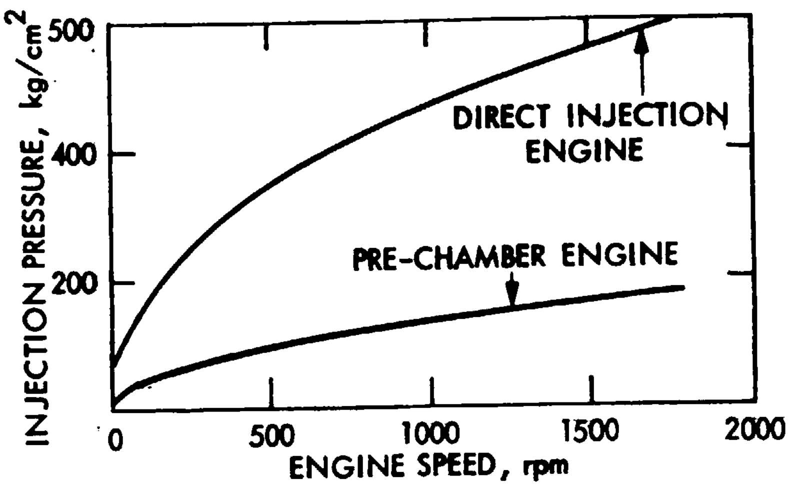


Figure 2.1-2. Comparison of Fuel Injection Pressures (Ref. 1)

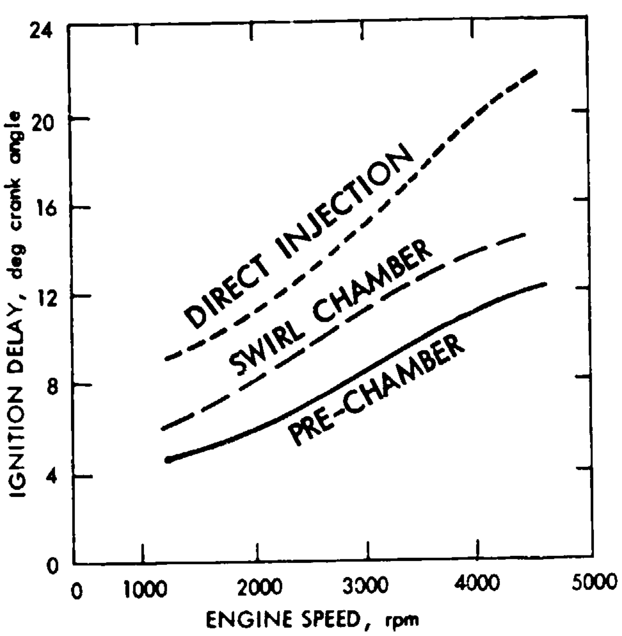


Figure 2.1-3. Effect of Chamber Design on Ignition Delay (Ref. 1)

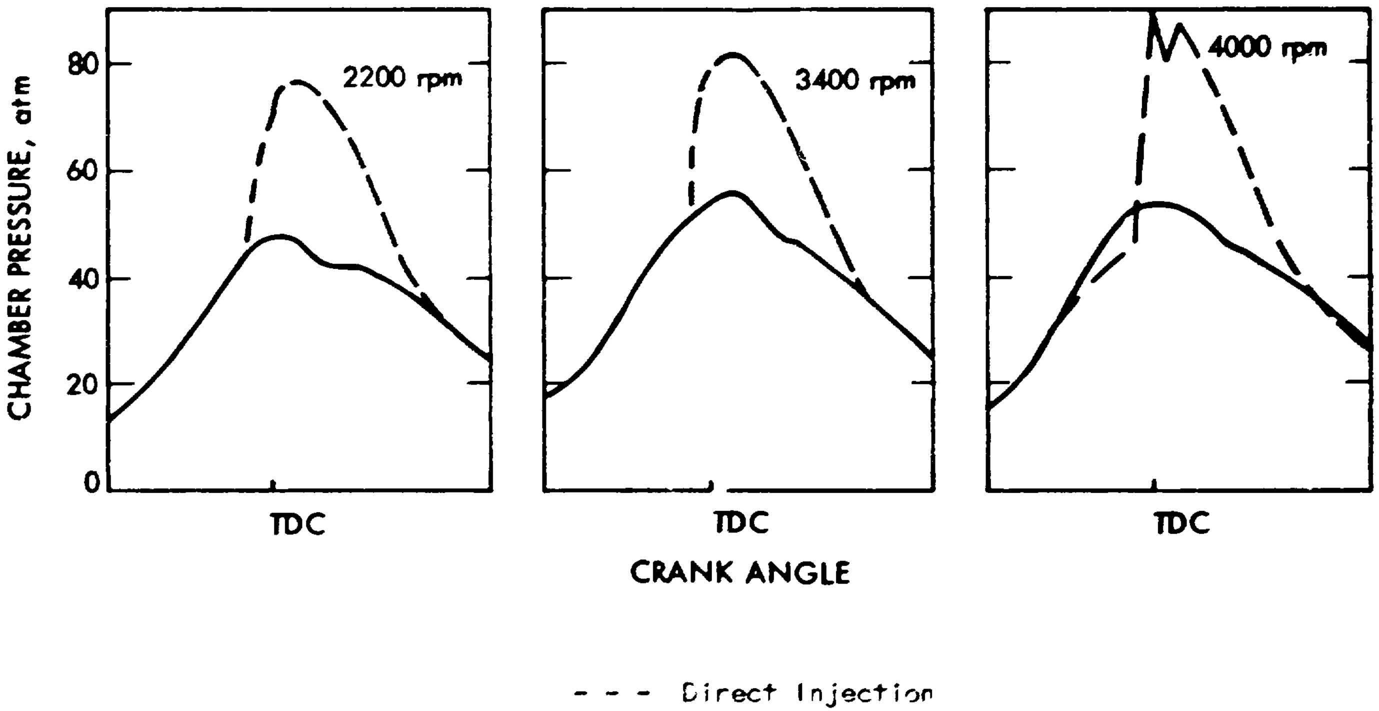


Figure 2.1-4. Comparison of Main Chamber Pressures (Ref. 1)

sary penetration distance and makes it possible to more accurately meter and control the fuel injection rate at high engine speeds. The swirl chamber concept was introduced years ago by Ricardo, and most automotive diesels use the swirl chamber.

As shown in Figure 2.1-5, the advantages of the pre-chamber and swirl-chamber are offset by internal pressure losses, between the main chamber and the pre-chamber due to flow restriction through the connecting throat. These losses are caused by the pumping of air and hot gases from one chamber to the other and, as shown in Figure 2.1-6, increase with engine speed. In terms of engine efficiency (Figure 2.1-7), the high speed, divided chamber diesel part load efficiency falls between that of the spark-ignition engine and the open chamber DI diesel engine, due to losses caused by internal pumping between chambers. At high engine speeds, the pumping losses essentially cancel the advantages of the pre-chamber diesel over the spark-ignition engine.

The problem of controlling fuel flow in high-speed diesel engines becomes apparent in Figure 2.1-8. Plotted as a function of crank angle are the fuel pump displacement rate (9) and the resultant system variables. These variables include nozzle needle lift (8) and orifice size, fuel flow rate (10) and droplet penetration (5 and 6), the absolute amount of fuel administered to the chamber (7), plus the resultant pressure in the pre-chamber (3). Comparing the fuel flow rate at the pump outlet (9) with the flow rate actually leaving the nozzle (10), the figure shows that a considerable lag and distortion of the flow-rate profile exists, which becomes more pronounced as engine speed increases. In this particular case, the pump feeds over only a 10° crank angle starting shortly before the top dead center (TDC), whereas the fuel injection period extends over almost 30° or 20° past the TDC. The interaction between components, liquid inertia and fuel line expansion and contraction due to pressure are primarily responsible for this.

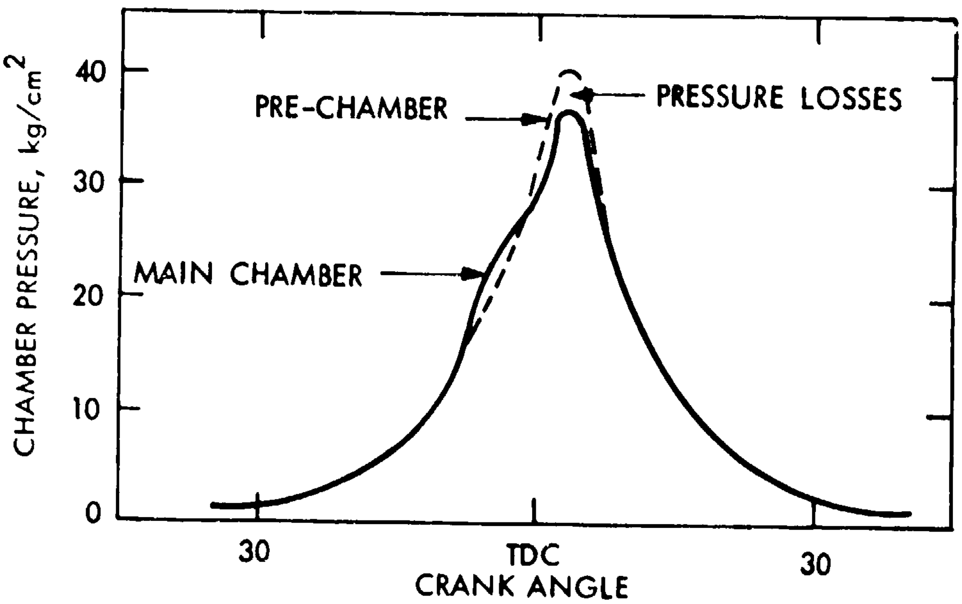


Figure 2.1-5. Comparison of Chamber Pressure Profiles (Ref. 1)

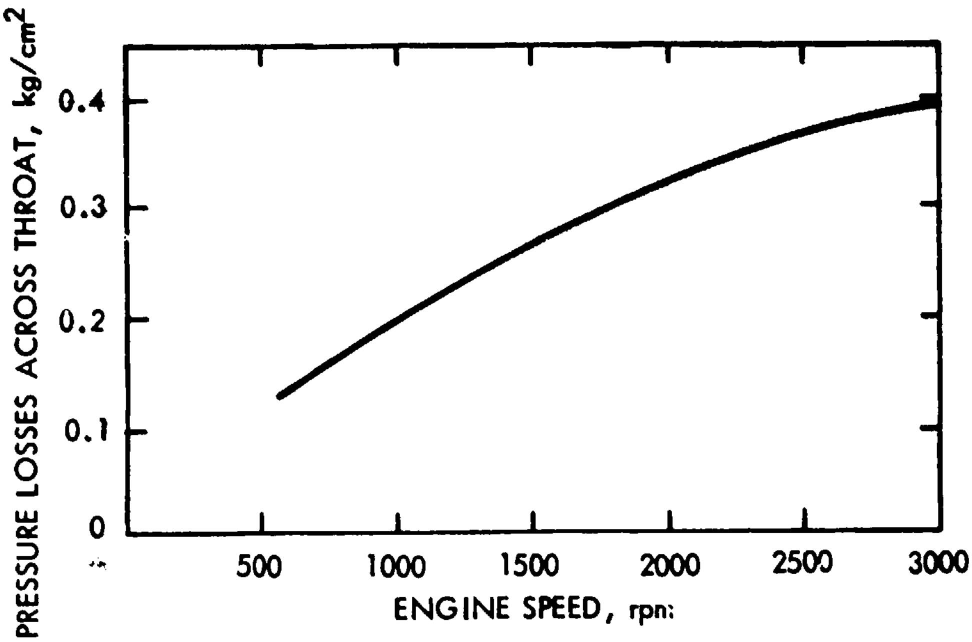


Figure 2.1-6. Effect of Engine Speed on Pre-Chamber Pumping Losses (Ref. 2)

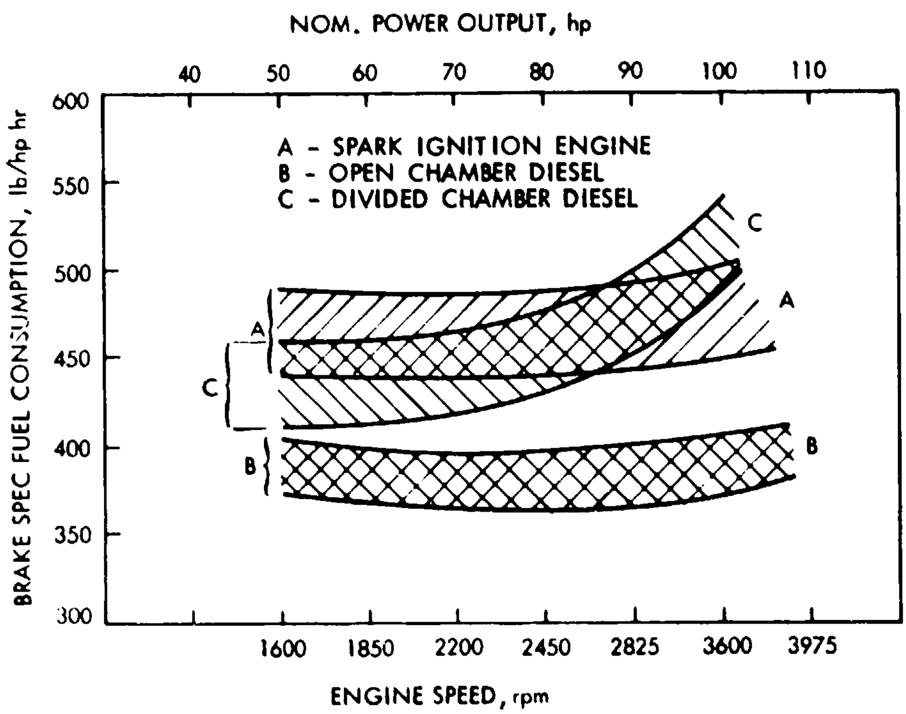
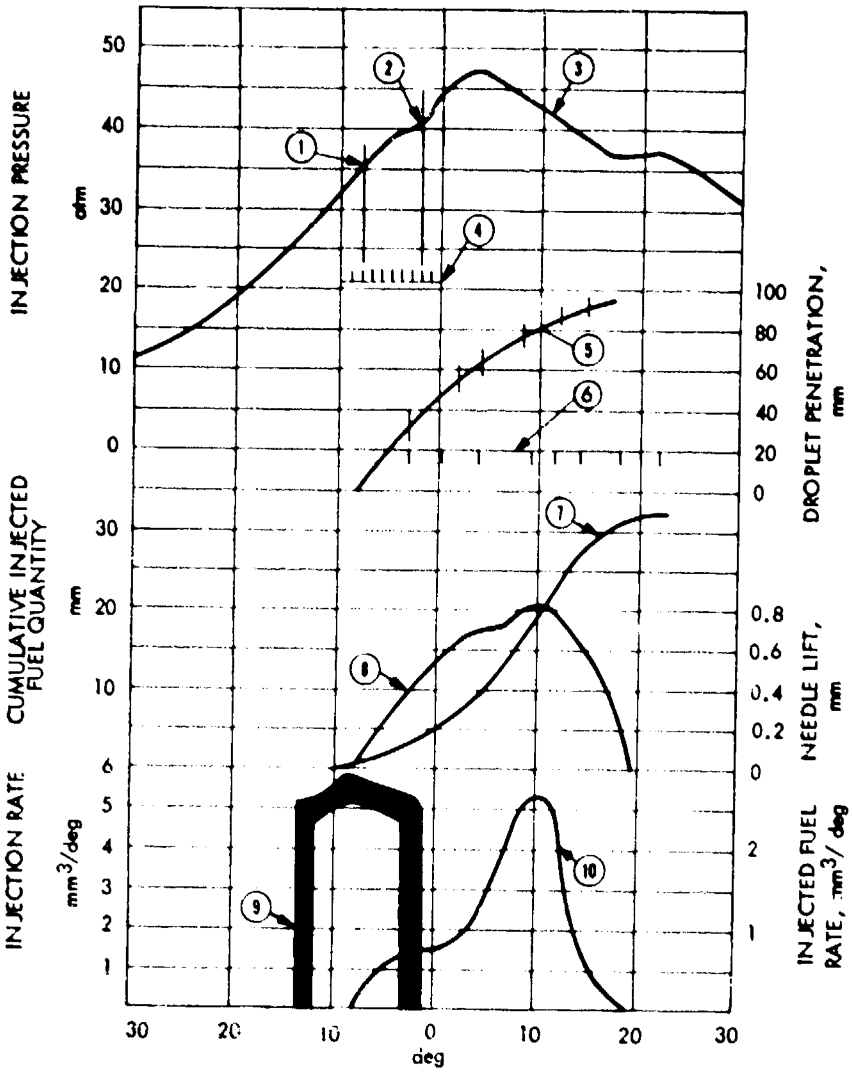


Figure 2.1-7. Comparison of Brake Specific Fuel Consumption (Ref. 3)



LEGEND:

1 - START OF INJECTION

2 - START OF IGNITION

3 - PRECHAMBER PRESSURE, Kg/cm^2

4 - IGNITION DELAY, deg

5 - DROPLET PENETRATION WITHOUT BALLPIN, mm

6 - DROPLET PENETRATION WITH BALLPIN, mm

7 - CUMULATIVE INJECTED FUEL QUANTITY, mm^3

8 - NEEDLE LIFT, mm

9 - PUMP DISPLACEMENT, mm^3/deg

10 - INJECTED FUEL RATE, mm^3/deg

Figure 2.1-8. Injection Characteristics of a Typical High-Speed Automotive Diesel at Maximum Torque (Ref. 1)

The figure also shows at (5) that approximately 15% of the total fuel injected has accumulated in the chamber before self-ignition will actually start. This is due to ignition delay. It is also the primary cause for the sudden initial heat release typical for diesel engines which, in turn, is the major cause for the knocking noise generated by diesel engines. Figure 2.1-9 shows that the improvement in air turbulence, mixing and reduced ignition delay brought about by the introduction of pre- and swirl chambers (B) and (C), has considerably flattened the heat-release profile as compared to the DI engine (A).

Although the internal stresses and friction in diesel engines are relatively high, conservatively designed diesels usually outlast Otto engines of the same category by a factor of up to 2. The sturdier construction, closer hardware tolerances, and increased accuracy, as well as the late injection of the fuel, are mainly responsible for the prolonged diesel life. The late injection of the fuel prevents film wash and oil dilution, one of the primary contributors to engine wear, particularly during cold starts. No tune-ups are required with diesels because all major operational functions (timing and fuel metering schedules), are designed into the system. However, the oil must be changed more often because of contamination by particulates generated during diesel combustion. In summary, maintenance and fuel costs are generally lower, and time between major overhauls is considerably longer for diesel engines, which offsets the higher initial cost of the diesel as compared to an Otto engine of the same displacement.

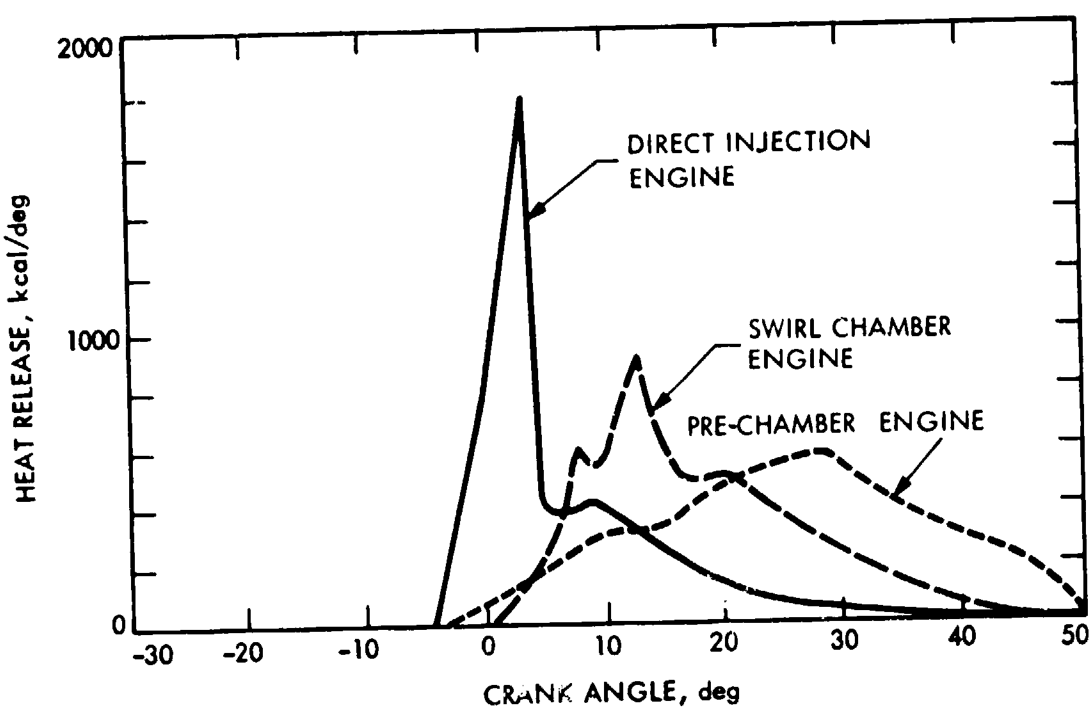


Figure 2.1-9. Heat Release vs. Crank Angle (Ref. 1)

Gasoline-diesel conversions such as the ones developed by VW and General Motors (GM)/Oldsmobile are compromised designs that share a variety of components and tooling with their gasoline counterparts. Although reinforced where necessary to cope with higher cylinder pressures, they are not expected to last as long as conventionally designed diesel engines. There is not sufficient road experience available at the present time to draw conclusions. Volkswagen does not claim that their diesels will outlast their gasoline counterparts. Most of the diesel Rabbits on the road have accumulated an average of about 60,000 miles without major complaints. General Motors/Oldsmobile diesel conversions have exhibited initial mechanical problems which have reportedly been remedied and cannot be considered typical for gasoline-diesel conversions.

An inherent problem with all diesel engines is that of cold starting. In order to reach the required self-ignition temperature with cold fuel and a cold engine, the compression ratio (approximately 22) built into small diesel engines is usually higher than desirable from the fuel economy standpoint alone. A heater element, or a so-called glow-plug, must be provided to preheat the entrapped air prior to engine start. Depending upon system capability and ambient temperature, 6 to 60 s of heating is normally required before the engine can be cranked up for start. More battery capacity is usually provided on diesel cars because of preheating and the higher starting torque required.

The diesel is better adapted for the application of supercharging techniques than the gasoline engine because the diesel does not have a fuel knock (pre-ignition) problem. Supercharged diesels have about the same power per unit displacement and unit weight as comparable gasoline engines, but much additional design and development effort is necessary to cope with the higher system pressures, and structural and thermal loads.

To date, all passenger-car diesel engines developed are of the four-stroke type. This does not mean that two-stroke engines do not have their place in the small automotive diesel category, as they do now among truck engines. The simple geometry of two-cycle combustion chambers, lends itself more favorably to the introduction of ceramic materials than the head configurations of four-cycle engines. The inherent problem with residual gases in two-cycle engines might be used to an advantage as an exhaust gas recirculation (EGR) system.

2.2 VEHICLE FUEL ECONOMY AND PERFORMANCE

Diesel engines are usually offered as an option in a car that will accept either engine with minor changes in the frame and suspension system. Therefore, most comparisons are made on the basis of diesel/gasoline options that have about the same displacement and box-volume for a given inertia weight. Diesel engine weight penalties are relatively small in relation to the vehicle inertia weight, and can be neglected where the essential differences depend primarily upon the drive-cycle. An infinite number of drive-cycle variations are possible. The extremes range from steady-state vehicle operation on a level surface with a thermally conditioned engine, to driving in frequent stop-and-go traffic with a cold engine.

Figure 2.2-1 compares the road load fuel economy of a 2-ℓ Otto/diesel engine option operating in high gear, showing a fuel economy advantage on the order of 100% or more over the gasoline engine at slow speeds. At high-speed full-performance driving, the fuel economy advantage over the diesel almost vanishes, because both engines run unthrottled. Flow pressure losses across the inlet valves and pre-chamber pumping losses then become the dominant factors contributing to the internal losses of both engines. As shown in Figure 2.2-2 the diesel's inherent part-load efficiency characteristic, which is considerably flatter than that of conventional gasoline engines, is mainly responsible for the diesel's excellent fuel economy at low vehicle speeds. In low-speed operations involving frequent stops and waiting times, as in city driving, the diesels low-idle fuel flow rate is a strong contributor to vehicle fuel economy. As can be seen from Figure 2.2-3, the idle fuel flow rate of the diesel is less than one third that of a conventional uniform charge gasoline engine, and less than one half that of a stratified charge gasoline engine.

The diesel's fuel economy advantages are apparent when comparing Environmental Protection Agency (EPA)-measured fuel economy data for highway and urban driving. Table 2.2-1 compares the EPA ratings of diesel/gasoline options for 1980 and 1981 model cars. Based on composite values, the diesel options exhibit a fuel economy gain of 24 to 57%, with a 38% average gain over the gasoline option. The same average fuel economy advantage is obtained, as shown in Table 2.2-2, if diesel/gasoline engine options are compared on the basis of sales-weighted average inertia weight. The sales-weighted

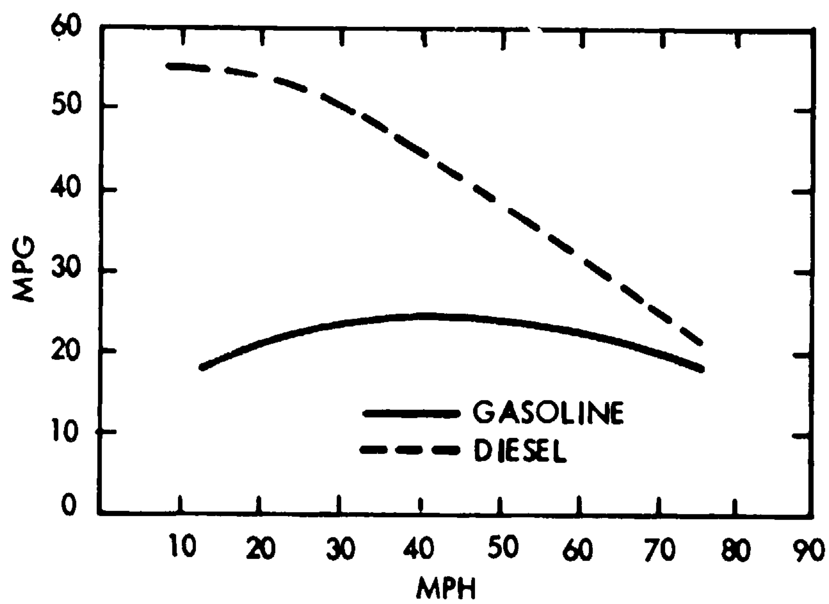


Figure 2.2-1. Comparison of Gasoline vs. Diesel Fuel Economy at Steady Road Load with a Standard-Size European Production Passenger Car (Ref. 3)

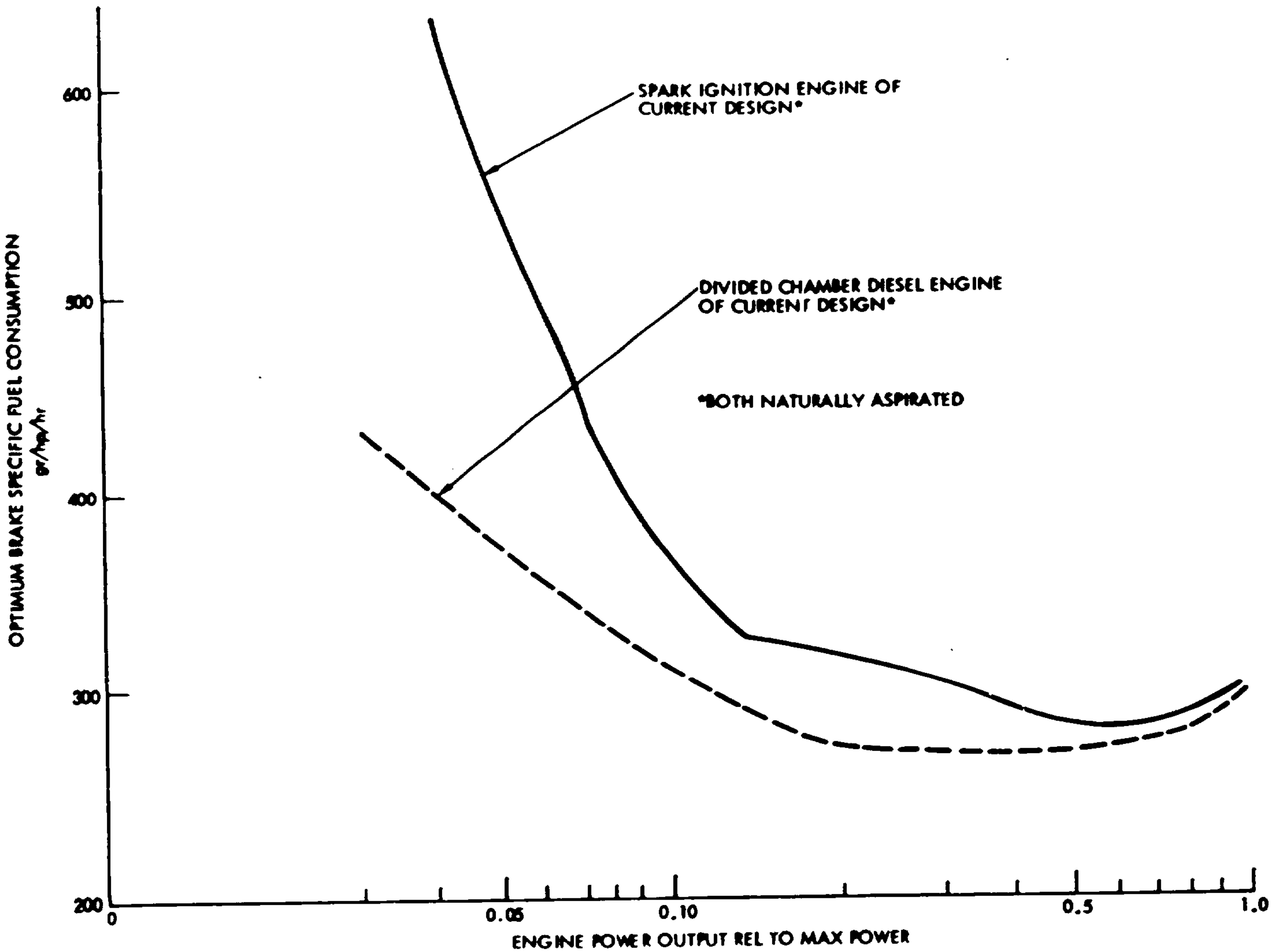


Figure 2.2-2. Comparison of Optimum Part-Load Brake-Specific Fuel Consumption (Ref. 4)

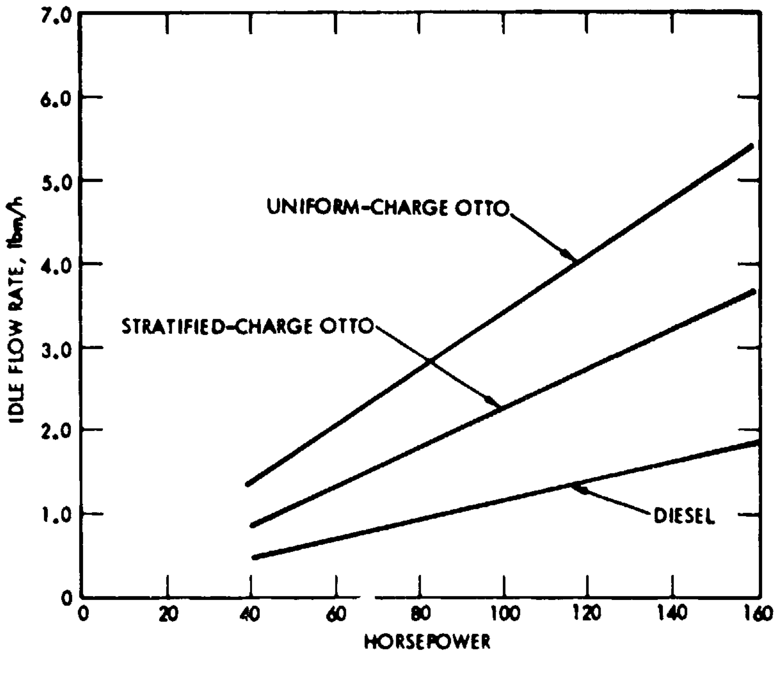


Figure 2.2-3. Comparison of Idle Fuel Flow Rates (Ref. 5)

comparison, however, suffers from the fact that the large cars have been more subject to fuel-economy oriented power reductions during recent years than the small cars that are already more fuel-efficient. Compared with the gasoline powered economy cars of the same weight class as shown in Table 2.2-3, the diesel average fuel advantage over the gasoline powered car is only on the order of 25%. This number is the present EPA consensus (Ref. 6) regarding diesel fuel economy advantages when compared with the 1980/1981-model gasoline engine mix currently on the road, including three-way catalyst controlled and fuel injected engines.

As can be seen from Table 2.2-4, the fuel economy advantages of the current-design diesel reduce drastically if compared to spark-ignition-engine powered cars that have the potential to be on the road by the mid 1980s. Improved and advanced engine systems that will compete with the diesel by then, in large numbers, are three-way-catalyst equipped and electronically controlled fuel-injected engines, cylinder cut-off engines, anti-knock controlled TC gasoline engines, and a general trend in gasoline engine design

Table 2.2-1. EPA Fuel Economy Comparison by Model Type Gasoline and Diesel Vehicles

Model, 1980[a]	Engine/Transmission	Composite mpg (City/Highway)	Difference, %[b]
Audi 5000	121 CID/MS Diesel	35 (27/43)	57
	131 CID/MS Gasoline	21 (17/30)	
Cadillac	350 CID/A3 Diesel	23 (20/30)	35
Seville/Eldorado	368 CID/A3 Gasoline	17 (14/22)	
Chevrolet C10	350 CID/A3 Diesel	23 (20/27)	28
Pickup	305 CID/A3 Gasoline	18 (15/19)	
IH Scout	198 CID Diesel/M4	22 (20/24)	38
	196 CID Gasoline/M4	16 (15/19)	
Mercedes Benz			
300 D	183 CID Diesel/A4	22 (23/28)	39
280 SE	168 CID Gasoline/A4	18 (16/20)	
Oldsmobile	350 CID Diesel/A3	26 (22/34)	
Cutlass	305 CID Gasoline/A3	19 (17/24)	37
	260 CID Gasoline/A3	21 (19/24)	24
Oldsmobile 88	350 CID Diesel/A3	26 (22/34)	
	307 CID Gasoline/A3	20 (17/25)	30
	260 CID Gasoline/A3	19 (17/23)	37
Peugeot 504	141 CID Diesel/A3	29 (28/32)	38
	170 CID Gasoline/A3	21 (19/25)	
Volkswagen	90 CID Diesel/M4	41 (36/49)	52
Dasher	97 CID Gasoline/M4	27 (23/35)	
Volkswagen	90 CID Diesel/M4	45 (40/50)	41
Rabbit	89 CID Gasoline/M4	32 (27/41)	
Volvo Station	145 CID Diesel/M4-OD	31 (28/38)	41
Wagon (1981)	130 CID Gasoline/M4-OD	22 (18/29)	

Average Model Type Difference = 38%

[a]Except where otherwise noted.

[b]$\text{mpg difference} = \frac{\text{Diesel-Gasoline}}{\text{Gasoline}}$

Data taken from EPA "Test Car List," 1980 Second Edition.

Table 2.2-2. EPA Fuel Economy Comparisons Gasoline and Diesel Vehicles, 1980 Model

Saleweighted Average mpg[a] by Inertia Weight Class						
	2250	2500	3000	3500	4000	4500
Diesel, mpg	45.4	41.1	32.6	26.7	26.4	24.2
Gasoline, mpg[b]	31.4	27.5	23.9	20.6	18.5	17.5
Improvement, %	45	49	36	30	43	38

Salesweighted Average Improvement[c] = 38%

Unweighted Average Improvement = 40.1%

[a]EPA Composite mpg.

[b]From Table C2 of SAE #800853, corrected to reflect gasoline vehicle only.

[c]Based on total (gasoline and Diesel) sales fractions within classes normalized to total sales in the weight classes listed.

Data taken from "Passenger Car and Light Truck Fuel Economy Trends Through 1980," J.D. Murrell, et al., SAE Paper #800853, June 1980.

to return to higher compression ratios. An improvement of engine part-load fuel efficiency characteristics approaching those of the diesel, as shown in Figure 2.2-4, are responsible for the reduced diesel advantage over gasoline engines. However, the diesel community is not resting on its laurels and concerted R&D efforts are underway to maintain the fuel advantage of the diesel over the gasoline engine. Major and prospective R&D effort continues in the direction of electronically controlled open chamber combustion, ceramic lining of combustion chambers to reduce cooling losses, and turbo-compounding, which adds up to a total improvement potential on the order of 40 % over the current design divided-chamber diesels. This work is described in more detail in Section 5.0.

Because the diesel operates generally at a higher air-to-fuel ratio, the naturally aspirated diesel produces approximately 30 to 35% less torque than gasoline engines of comparable size. Therefore, diesel cars are usually geared lower to compensate for the lack of acceleration, thus reducing the

Table 2.2-3. Fuel Economy Comparisons for the Best of 1980 Model Gasoline and Diesel Cars, Highest mpg by Inertia Weight Class

	Weight Class					
	2250	2500	3000	3500	4000	4500
Diesel, mpg[a]	47.4	41.1	32.6	31.6	27.3	24.2
Gasoline, mpg[a]	42.4	35.4	28.1	22.3	21.7	18.2
Improvement, %	12	16	16	42	26	33

Average Improvement = 25%

[a]EPA Composite mpg.

Data taken from "Passenger Car and Light Truck Fuel Economy Trends Through 1980," J.D. Murrell, et al., SAE Paper #800853, June 1980.

top speed by about 10%. Supercharged diesel cars exhibit driveability comparable to that of naturally aspirated gasoline engines of the same displacement. By the mid 1980s most diesel passenger cars marketed will probably be TC. Small displacement diesels too small to be TC will possibly be supercharged for acceleration and passing, using mechanically driven or shock-wave (comprex)-type compressors, and R&D work in this direction is currently in progress.

2.3 EMISSION

Diesel-powered cars meet 1980 emission standards without the use of catalytic converters and exhaust gas recirculation. They have hydrocarbon (HC) and NO_x emissions comparable to open loop controlled catalyst-equipped spark-ignition powered cars, with carbon monoxide (CO) emissions generally lower. However, in addition to the regulated gaseous pollutants, diesel engines emit particulates, smoke, oxides of sulfur, odor and aldehydes. Of particular concern are the relatively large quantities of particulate matter and absorbed organic compounds emitted by diesel engines, some believed to be carcinogenic.

Table 2.2-4. The mpg Advantage of a TC Diesel Engine of Current Design Over Contending Current and Advanced Spark Ignition Engines (Ref. 4)

Contending Spark-Ignition Engine Concepts	Diesel Advantage Over Contender, %[a]	Remarks
Conventional low compression (8.5:1) engine fuel-injected	53	1.7 ℓ Audi
Variable displacement (50/50) cylinder cut-off engine	40	Modified Porsche 924
Advanced high (12.5:1) compression engine	38	Porsche top (experimental 924 engine)
Variable inlet valve partially throttled engine	22	Analysis
Variable inlet valve/ variable compression ratio unthrottled engine	7	Analysis

[a]Composite Federal Cycle with 60 hp, 2400 lb reference car.

The emission standards projected for the future by the EPA (Table 2.3-1) provide for a limitation of particulate emissions to 0.6 g/mi for the 1981 model, and a further reduction of that level to 0.2 g/mi for the 1983 model, as well as a reduction of the NO_x standard from 2 to 1 g/mi in 1981, and to 0.4 g/mi from 1983. As discussed in more detail in Section 4.0, compliance with such standards is an unresolved problem. Figure 2.3-1 shows that the problem is compounded by the fact that, for a given inertia weight, the technical measures which reduce NO_x and particulates are not mutually compatible. In the opinion of most automobile manufacturers and diesel experts, a NO_x standard of 1.5 g/mi can be met with existing diesel engines by means of injection and timing refinements without the use of a special emission control system. Compliance with a NO_x standard of 1.0 g/mi or less will require substantial amounts of EGR, with attendant increase in HC and particulate emission, plus degradation of fuel economy and durability, as well as increased cost. The general concensus is that a 0.4 g/mi NO_x standard will not be attainable at all with current predictable technology.

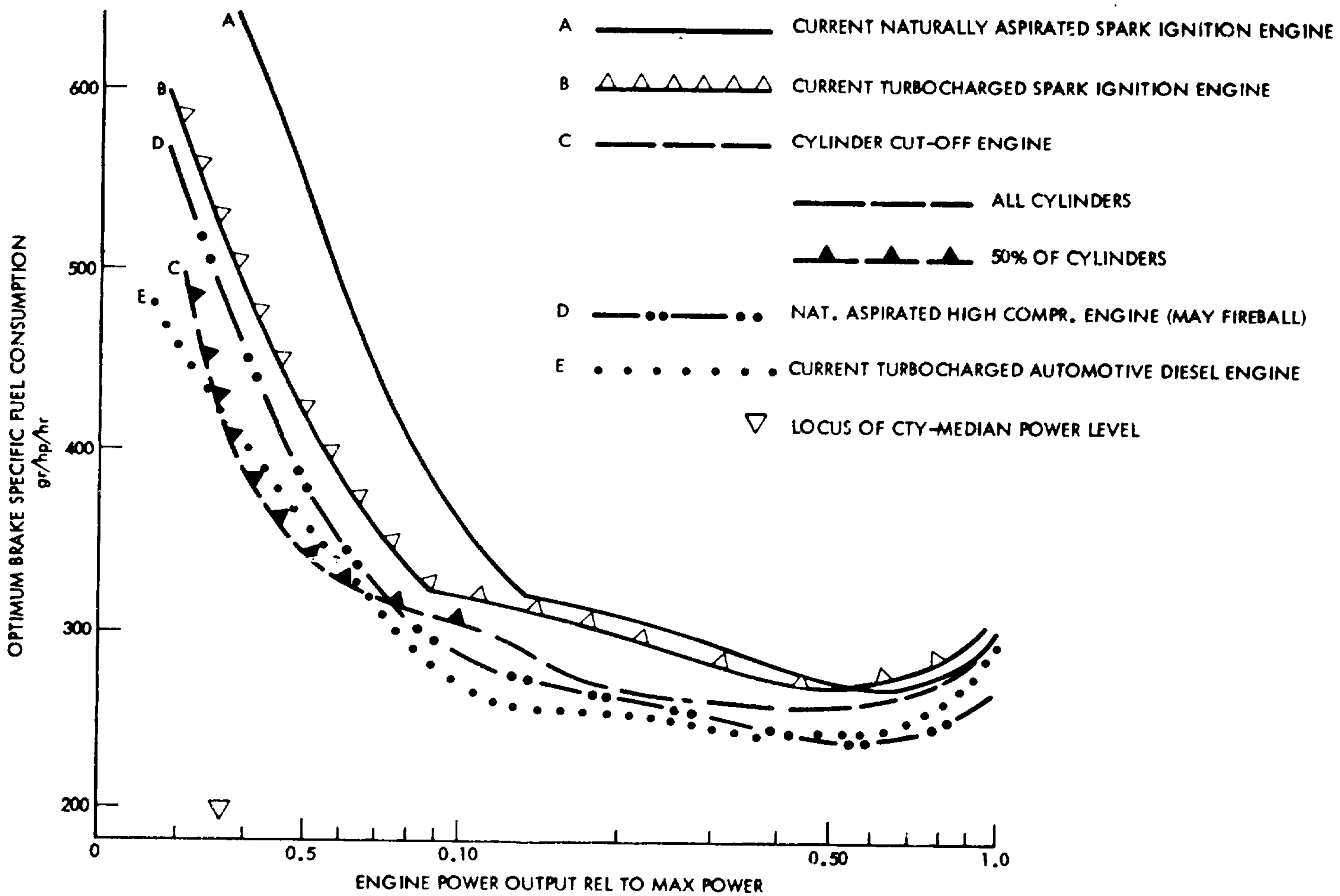


Figure 2.2-4. Comparison of Optimum Engine Part-Load Specific Fuel Consumption (Ref. 4)

Table 2.3-1. Current and Projected Standards for Gaseous and Particulate Emissions (Ref. 7)

Model Year	Allowable Pollutants, g/mi							
	HC		CO		NO_x		Particulates	
	Cal.	Other States	Cal.	Other States	Cal.	Other States	Cal.	Other States
1979	0.4	1.5[b]	9.0	15.0	1.5	2.0	--	--
1980	0.39	0.41[c]	"	7.0	1.0	"	--	--
1981	"	"	3.4(7.0)[a]	b	1.0(1.5)[a]		--	0.6
1982	"	"	7.0	"	0.4(1.5)[a]	"	0.4(0.6)[a]	0.6
1983	"	"	"	"	0.4	"	"	0.2(0.6)[a]
1984	"	"	"	"	"	"	"	"
1985	"	"	"	"	"	"	0.2	"

[a]EPA waivers
CARB waivers with 100,000 mi emissions contr. system warranty.

[b]Standard based upon non-methane hydrocarbon emission.

[c]Standards including methane.

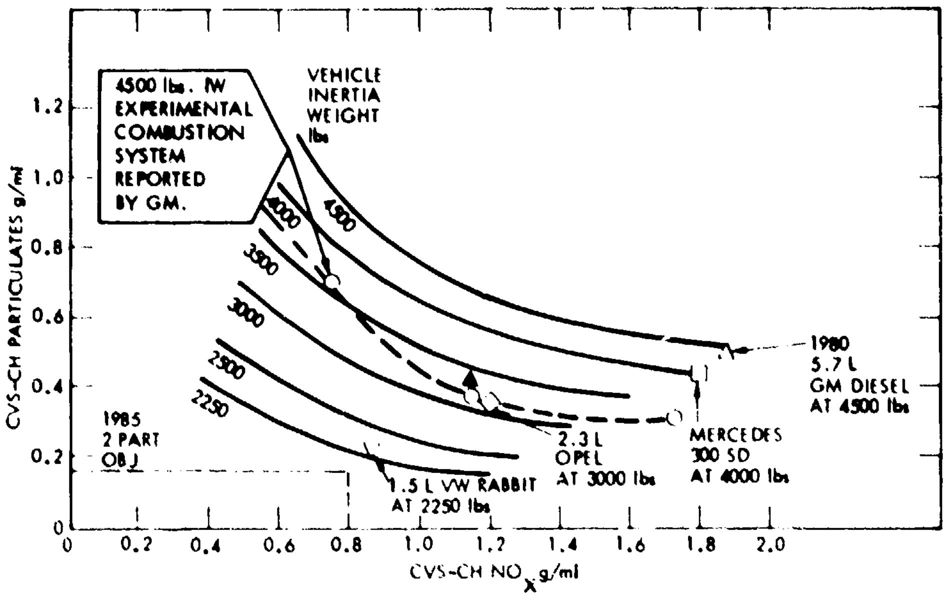


Figure 2.3-1. Particulate Emissions vs. NO_X Emissions for Light-Duty Diesel Vehicles with Inertia Weights Ranging from 2250 lb to 4500 lb

To date, only the 1.5-ℓ four-cylinder VW Rabbit diesel, the smallest of all, can meet the 1981 particulate standard of 0.6 g/mi and 1.0 g/mi NO_X by a narrow margin. The others, which are larger, will have problems meeting the EPA proposed schedule (Table 2.3-1) for NO_X and particulate emissions. General Motors, the producer of the largest diesel cars marketed in the United States joined by Mercedes Benz (MB), Peugeot, and VW, asked the EPA in 1979 to grant a 4-yr waiver from the 1981, 1.0 g/mi NO_X standard to a 1.5 g/mi level, and reprieve from the 0.6 g/mi standard for particulates starting in 1981. Under the 1977 Clean Air Act, such a waiver is available if the auto maker can convince the EPA that a waiver:

(1) Is necessary to permit the use of diesel engine technology.

(2) Does not endanger public health.

(3) Will result in significant fuel saving.

(4) That the technology used has potential for long-term air quality benefits.

In response to an extensive study conducted by GM (Ref. 8) and other similar studies that substantiate the above requirements and the need for more development time, the EPA has delayed the imposition of the 0.6-g/mi standard for particulates until 1982. It has also granted a 2-yr waiver (instead of 4) from the 1.0-g/mi NO_X standard to a level of 1.5 g/mi. The California Air Resource Board (CARB) has approved of this waiver for cars marketed in California provided the manufacturer warranties the proper functioning of the emission control and associated fuel systems for 100,000 mi.

For VW, which meets the 1.0-g/mi NO_x standard by a narrow margin with the 1.5-ℓ Rabbit diesel, the 1.0-g/mi NO_x standard was was relaxed only to 1.3-g/mi for the Rabbit, to 1.4 g/mi for the Dasher, and to 1.5 g/mi for the five-cylinder Audi diesel car. All others, regardless of size, have to adhere to the 1.5-g/mi NO_x standard during the waiver period. The current waiver situation may become statutory until new ways to simultaneously reduce NO_x and particulate emissions can be found.

2.4 FUEL AVAILABILITY AND COST

Diesel fuel is presently available from more than 10,000 stations throughout the United States along the major highways and in metropolitan areas. Most of these are truck stops, as only about 5% of the gasoline service stations carry diesel fuel. According to the number shown in Table 2.4-1, the price of diesel fuel has risen during recent years along with that of gasoline with an average difference remaining at 13%.

Indications are that the price of diesel fuel is possibly going to rise beyond that of gasoline in the future. The general increase in fuel prices early in 1979 was the result of a world-wide shortage of crude oil brought about by the OPEC "embargo." Since then the price of diesel fuel has risen at a faster rate than that of gasoline because of the U.S. diesel fleet growth during that period of time. The trucking industry has made recent demands for special diesel fuel allocation and price control. If these demands are met, the existing shortage in the private sector will be aggravated, and the pump price of diesel fuel being sold to private consumers will be strongly impacted.

In addition to the relationships between supply and demand, the price of diesel fuel will also be dictated by the production cost at the refinery. According to studies conducted by Amoco and Mobil Oil Co. (Ref. 9), refinery costs are now primarily from gasoline production but will shift more to diesel production. From a specific value of gasoline to distillate ratio and below, diesel production is more expensive because it requires more energy for desulfurization and hydrocracking. Figure 2.4-1 shows, for example, the change in gasoline to distillate (G/D) ratio required for a shift from gasoline to diesel fuel, assuming that the percentage of all distillates remains unchanged. As indicated by the data point at a G/D = 0.9, the fraction of diesel fuel that can be obtained at a given G/D ratio is less if the percentage of other distillates such as jet fuel are also increased in the future. Refineries operating at a typical gasoline-to-distillate ratio of 1.7 can increase their output of diesel fuel by 10% with minor changes in existing systems. Beyond that, major system changes are required. A maximum shift to diesel fuel on the order of 40% is said to be feasible in a refinery which is optimized for diesel fuel production.

In any case, substantial investment is required, and must be recovered. For example, Figure 2.4-2 shows the effect of refinery change in the direction of more diesel fuel, normalized to the price of diesel fuel (excluding taxes) for a refinery operating at a normal gasoline-todistillate ratio of 1.7. As can be seen, pricing policy has a strong influence. In case A,

Table 2.4-1. Fuel Price Comparisons of Unleaded Gasoline and Diesel Fuel #2 (Cents Per Gallon Retail with Tax)

	Unleaded Regular Gasoline[a]	Diesel Fuel #2[b]	Difference[c]
1976	61.4	52.7	8.7 or 14%
1977	65.0	57.3	7.7 or 12%
1978	67.0	58.2	8.8 or 13%
1979	90.3	81.4	8.9 or 10%
1980	126.9	109.2	17.7 or 14%

Average Difference 13%

[a]Mix of "Full Serve" and "Self Serve" from Reference 11.

[b]Based on adjusted (see text) retail price (ex tax) from Reference 10, plus a constant 15 per gallon for state and Federal taxes.

[c]% Difference = $\frac{\text{Gasoline-Diesel}}{\text{Gasoline}}$

(Figure 2.4-2A), the price of gasoline is artificially kept constant and the diesel price is adjusted as necessary to maintain refinery revenue. In case B, the diesel price rises in proportion to incremental cost, and the gasoline price is adjusted as necessary to maintain refinery revenue. If gasoline prices are allowed to drop according to that fuel's fair share of the production cost, then diesel prices will be forced up even more than shown in Figure 2.4-2B. This would severely dampen consumer interest in diesel cars.

If it is important from the overall energy standpoint, a certain regulation of the fuel prices would be necessary to keep the consumer interest in diesel cars alive. In France (Ref. 11), for example, the price of diesel fuel and gasoline is currently regulated by taxation so that the diesel fuel price remains about 1/3 below that of gasoline. This has drastically increased the use of diesel cars by a factor of four during the last 5 years. With further dieselization, this differential in price is expected to decrease and to level off at 10 to 15%.

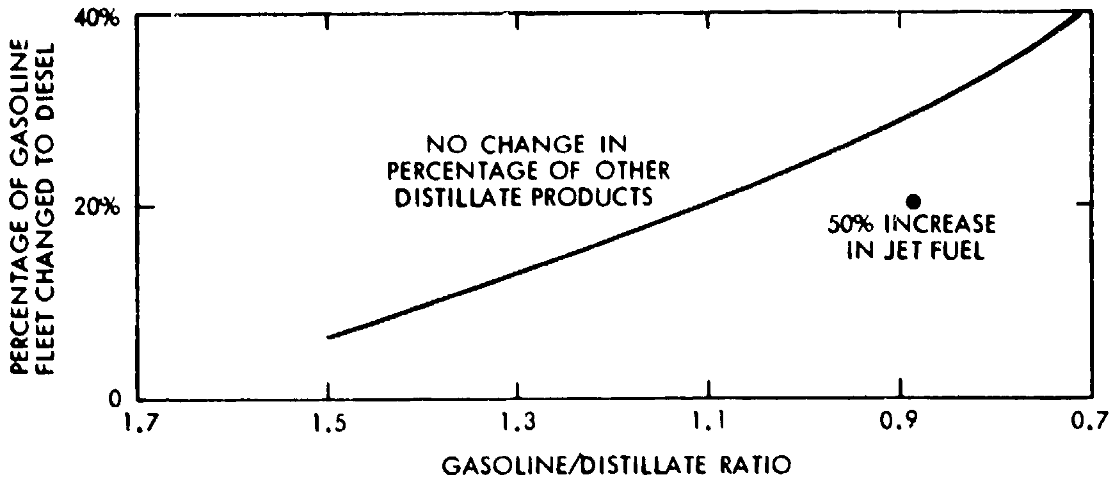
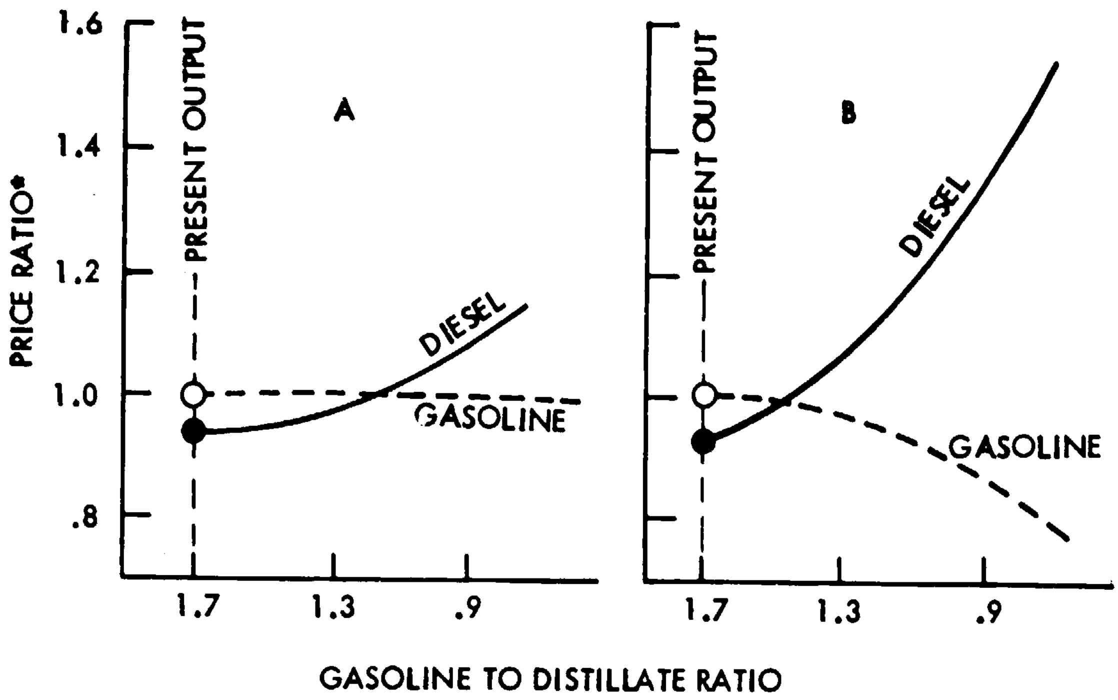


Figure 2.4-1. Effect of Fleet Change from Gasoline to Diesel Fuel on Distillate-to-Gasoline Ratio (Ref. 10)

Another factor closely related to fuel cost is that of fuel quality. Complaints have been voiced recently in press reports (Ref. 13) about the large amounts of water found in diesel fuel delivered at retail pumps. In an isolated but extreme case, 24-gal of water were found in a 27-gal tank. This is not a representative case but the general situation seems critical enough to warrant further investigation and corrective action. Current Oldsmobile passenger car fuel systems are designed to handle up to 10% water. If the filter is not effective, small amounts passing through the fuel pump and nozzles can result in fouling, which will result in costly repair bills.

The oil companies claim that diesel fuel with low water content can be delivered to jobbers and retailers, but that water is taken on during transfer between storage facilities. Because it is heavier than diesel fuel, the water settles to the bottom of the storage tanks, and when supplies become scarce, distributors pump deeper into the tanks than they normally do. Another problem is that many stations are switching from low demand fuels such as premium gas to diesel fuel, and do not properly flush storage tanks before use. The use of old, leaking tanks that allow ground water to seep in is also a problem.

The three major European diesel car producers, MB, Peugeot, and VW, are said to downplay the water content problem because it might discourage prospective buyers. According to GM, present diesel engine systems are designed to cope with a water content in the fuel of up to 10%. Beyond that, corrective and preventive measures must be taken, from the main storage facility to the engine nozzle. The cost to accomplish this task must be passed on to the diesel customer. At the present time, GM recommends buying diesel fuel at truck stops, rather than in gasoline stations. Truck stops have a fast turnover, and truckers do not tolerate water in their fuel.



*prices in relation to present price of gasoline at the refinery, less taxes

Figure 2.4-2. Effect of Gasoline-to-Distillate Ratio on Gasoline and Diesel Fuel Pricing (Ref. 12)

Option A: Gasoline Price Constant, Diesel Price Adjusted to Maintain Refinery Revenue

Option B: Diesel Price Rises in Proportion to Incremental Cost. Gasoline Price Adjusted to Maintain Refinery Revenue

Since mid-1980 (Ref. 14) GM-diesel owners can have a water detection device installed in their vehicles. It is essentially an in-tank fuel strainer/water separator combined with a fuel-guage type float that flashes a lamp whenever more than 4-gal of water have collected in the tank. General Motors advises that you hurry to the nearest service station and syphon the water from the tank, using "convenient means" when this occurs.

2.5 GENERAL ECONOMIC ASPECTS

The reasons that a private consumer decides to buy a diesel-powered car are many and change with time. Until 1974 (Table 2.5-1), superior fuel economy, low maintenance and durability were the primary motivating factors. The concern of a possible gasoline shortage was a relatively small contributor.

Today, in contrast to the ratings shown in Table 2.5-1, the primary motives for buying a diesel car are: (1) freedom from concern over the gasoline shortage and the ever-increasing price of gasoline, and (2) maintaining ownership of a large and convenient luxury car by switching to a larger diesel-powered vehicle, which also has acceptable economy of operation. Group (1) primarily represents the buyers of small diesel-powered cars such as the VW Rabbit and Peugeot, and group (2) represents the buyers of mid-to-large size diesel-powered cars, such as the Mercedes 300D, the Oldsmobile Cutlass and the Cadillac Eldorado. Also, many of the new buyers are diesel buffs, or are merely intrigued by new and innovative engines and cars. Because of the interest of buyers other than the very economy-minded, the total number of diesel cars sold in the United States had tripled during the 1977-1978 time period and doubled during the 1978-1979 period. Optimistic predictions are that by 1985, 25% of all new cars marketed in the United States would be diesel-powered to meet expected customer demands (Ref. 10). The current market share of passenger car diesels sold in the United States is on the order of 3-1/2%.

The seemingly bright future of the diesel car is overshadowed by a number of very uncertain economic factors other than consumer demand. These factors will dictate the future diesel market trend, and will possibly inhibit the use of privately-owned diesel cars in the United States and in other free world countries. Of primary concern are the cost and energy aspects associated with a switch from gasoline to diesel fuel, and the controversial nature and technical feasibility of future emission standards for NO_x and particulates.

According to analysis (Ref. 9), taking the higher mpg capability of diesels and the higher energy content of diesel fuel into account, the total energy savings associated with the use of a diesel-powered vehicle is on the order of 11%. Additional savings will be realized at the refinery when switching from gasoline to more diesel fuel. For example, by reducing the gasoline-to-distillate ratio from the existing 1.7 to 0.9, 10% less energy would be used in the refinery relative to the energy contained in the additional diesel fuel produced. Adding this 10% to the above indicated 11% would bring the total energy savings to about 20%. A nationwide change from a gasoline/distillate ratio of 1.7 to 0.9 would provide enough diesel fuel to convert 20 to 30% of the gasoline fleet to diesel, depending upon

Table 2.5-1. How and Why Owners Decided to Purchase a Diesel-Powered Car

	Mercedes		Peugeot
	240D	300D	
Overall Economy	34%	30%	34%
Fuel Economy	20	25	30
Low Maintenance	12	16	19
Availability of Fuel During Gasoline Shortage[a]	7	14	7
Durability of Diesel Engine	17	10	7
Prior Ownership of Diesel Automobile	8	8	4
Engineering/Engine Design	3	7	3
Prior Experience with Non-Automotive Diesel Engine	6	5	7
Clean Emissions	4	4	8
Reliability of Diesel	7	4	5
Recommended by Friend/Relative	7	4	4
Energy Crisis[a]	7	3	6
Pollution-Control Devices Required on Gasoline Engines	3	3	7
Lower Cost of Diesel Fuel	3	2	8
Wanted to Try It/Experiment	3	2	3
Suits My Needs	3	---	3
All Other Reasons	31	9	24

[a]How a respondent would distinguish between these two questions was not explained.

Sources: J.D. Power & Associates, 1974, 1975.

the amount of distillate converted to jet fuel. For a conversion level below a ratio of 0.9, energy savings would drop sharply because hydrogen cracking would become necessary to produce more diesel fuel.

Although the net savings of automotive petroleum energy that would result from a 20 to 30% conversion of the gasoline fleet to diesel is only on the order of 4 to 6%, such a conversion would still be of interest from a national energy standpoint. It is questionable, however, whether consumer interest in diesel cars would keep up with the trend of fuel prices. The price trend shown in Figure 2.4-2B would definitely be a deterrent to buying a diesel car. Fuel prices would have to be regulated as shown in Figure 2.4-2A (or similarly) to favorably balance the demand for gasoline and diesel cars from the total energy standpoint.

A study conducted by the National Highway Traffic Safety Administration in 1977 (Ref. 10) arrived at essentially the same conclusion. But it was postulated that diesel penetration for the major auto producers will be about 5% in 1981, and will grow at a rate of 5% per year, reaching 25% by 1985. Assuming a 10-yr lifetime for diesel cars, this would constitute 7.5% of the total fleet by 1985, which is well within the limits that could be supported by existing refineries without major system changes.

In addition to energy considerations, the emission issue will be the primary factor in determining the fate of the diesel car. The maximum size of the diesel fleet will likely be determined by the emission levels that can be tolerated nation-wide, or perhaps in a given area. The latter would mean that the growth of the diesel fleet would be limited more in high density traffic areas than in the open country. In metropolitan areas, a curtailing of the private diesel fleet in favor of taxicabs and other commercial light duty vehicles is quite possible. Diesel cars may be completely banned in California because of the extremely stringent emission standards and accelerated time-schedules, which exceed those proposed by the EPA for the other 49 states (Table 2.3-1).

An economic factor which strongly affects consumer choice is the higher price of the diesel car. As mentioned earlier, diesels are more expensive than gasoline engines because they work under more demanding conditions (high operating pressures) requiring closer tolerances. Engine structural weights are increased to cope with higher stresses and larger amounts of metals are used to produce an engine. Diesel injection systems are high-precision mechanical devices that are more costly to make than carburetion and spark ignition systems. Diesel engines are made in smaller numbers, and brackets and frame changes are usually required to install the diesel option.

At the present time, the base price of a diesel option is about 10 percent above the price of the base gasoline engine-powered car. According to the common relationship between production cost and volume, the price of the diesel option should come down with increasing penetration and greater production volume, although this is very uncertain because the actual consumer price of diesel cars is greatly affected by supply and demand at the retail level. With diesel cars currently being in short supply, it is almost impos-

sible to buy a diesel from a retailer at the producer's suggested retail price. Dealers take advantage of this situation by loading diesel cars with a variety of costly options and dealer-installed accessories, many of which are not at all attractive to the economy-minded buyer.

Considering the total picture, the economic future of the privately owned diesel car is highly sensitive to a variety of factors, many of which are not predictable. This makes a long term market analysis less meaningful at this time. Those factors that are known optimistically suggest significant growth of the diesel market in the U.S. by 1985. These known factors are in essence:

(1) Improved fuel economy.

(2) The need to meet legislated Corporate Average Fuel Economy (CAFE) guidelines.

(3) The desire to continue with the production of family size cars.

(4) The costs and problems associated with meeting future fuel economy and emission requirements with gasoline engines.

(5) The increase in miles driven per barrel of crude oil that can be achieved with increased diesel penetration.

(6) The potential of meeting future stringent emission standards without the use of critical materials, which are now needed for the catalytic controlling of exhaust emissions.

SECTION 3

DEVELOPMENT STATUS AND BACKGROUND

3.1 BACKGROUND AND SCOPE

Diesel engines for light and medium duty vehicles are currently produced by Mercedes Benz, Peugeot, Perkins, Deutz, Opel, Fiat and Volkswagen in Europe; by Nissan, Toyo Kogyo, and Isuzu in Japan; and by General Motors/Oldsmobile in the United States. Up to this point in time (1981), only Mercedes Benz, Peugeot, Volkswagen, and General Motors offer diesel options with selected lines of cars in the United States. Renault, Fiat, BMW, and Volvo will also offer diesel options in Europe and in the U.S. in the near future. Ford, American Motors, and General Motors are planning to install diesels made in Japan by Toyo Kogyo and Isuzu into selected lines of subcompact cars to reserve a place in the small diesel market for the 1980s. Japanese car producers such as Datsun and Toyota will have diesel options for selected lines of passenger cars and mini-trucks starting with the 1981 model.

Because the diesel scenario is very complex and changes rapidly, the following discussion will concentrate on the current and well known production lines of selected producers, which are considered most representative of the current state-of-the-art in their specific category, and are of primary interest for the U.S. diesel market. Listed in order of market penetration, these are Mercedes Benz, Peugeot, Volkswagen, Oldsmobile, and Bavarian Motorworks/Steyr. This group is representative of two entirely different schools of thinking in regard to engine development: (1) engines that are designed from scratch to be nothing but a diesel (Mercedes Benz, Peugeot), and (2) diesel conversions that are based upon the extensive utilization of existing gasoline engine technology, production tooling, and engine components (Volkwagen, Oldsmobile). This group also comprises three of the most distinctive design features found in diesel engines - the pre-chamber (Mercedes Benz), the swirl chamber (Peugeot, Volkwagen, Oldsmobile), and the direct injection open chamber (BMW).

The case histories of Mercedes Benz and Volkswagen are described in more detail than was originally intended within the scope of this report. However, this was necessary to understand the technical difficulties, the effort, and the time frames associated with the development of automotive diesel engines.

3.2 MERCEDES BENZ

The first diesel passenger car (the 260 D) was introduced by Mercedes Benz (MB) in 1936. Since then, the line of diesel cars developed by MB has undergone a number of fundamental design changes. All MB engine models have one important design feature in common - the pre-chamber which has undergone a variety of changes and modifications without abandoning the original design concept.

Despite the success of the Ricardo swirl chamber which has been adopted by most other automotive diesel developers, MB has consistently adhered to the pre-chamber concept (Figure 2.1-1B) which it regards as the best compromise from the standpoint of operational smoothness, noise and turbocharging. Mercedes Benz claims that over a wide part of the operating range, the prechamber system makes possible an almost ideally constant combustion pressure, which is of primary importance with regard to noise abatement. With turbocharged engines in particular, mechanical loads and stresses on vital engine components are reduced. The MB pre-chamber exhibits lowest HC emissions, and NO_x emissions are marginally higher, when compared to a swirl chamber, (Figure 2.1-1C) under identical vehicle conditions and identical engine design. The slight advantages of the swirl chamber are not considered sufficient by MB to be of decisive importance in complying with future NO_x emission standards.

Table 3.2-1 shows the change of engine displacement, speed and performance over the years, which resulted from a zealous effort to improve diesel driveability and consumer acceptance. Since the introduction of the prechamber with the 260 D and 170 D models, the first major step to a modernized diesel was the change from conventional pushrod overhead valve design to an overhead camshaft as in the 190 D model (OM 621 engine) in 1958. The OM 621 engine then became the baseline for MB's diesel production program, and in its last stage is represented by the OM 616, a 2.4-ℓ engine having a bore of 91 mm. The gradual increase of engine displacement from the 1.8-ℓ 180D through the 2.4-ℓ 260D was accomplished without major changes in basic engine design. Of prime importance was a proprietary cooling slot arrangement between adjacent cylinders, which allows for an extremely narrow spacing of the cylinders. With the OM 616/240D, the performance capability of the MB four-cylinder line of engines had reached its limits of speed and displacement.

As discussed in Section 2.5, the sudden upswing in diesel sales because of the 1973 oil crisis and demands for more power and better driveability called for a further increase in engine displacement. Instead of going to a new six-cylinder design, MB added one more cylinder of the same bore and stroke to the OM 616 engine (Table 3.2-2). This represented the best compromise in terms of weight, size, investments and production cost, as compared to designing a new six-cylinder engine. The OM 617 was the first five-cylinder engine ever installed in a passenger car and was initially greeted with skepticism, although a five-cylinder in-line design has long been in use in stationary, marine, and truck engines. Careful engine balancing, a soft and carefully dampened engine suspension, and the installation of a torsion damper on the front end of the crankshaft have resulted in a vibrational characteristics similar to a six-cylinder engine. The engine was marketed for the first time in 1974 in the 300 D models, together with OM 615 and 616 four-cylinder engines which were then produced to power the 200D, 220D, and 240D lines.

In spite of the many uncertainties regarding the future of diesel cars, Mercedes Benz decided in 1976 to develop a more powerful turbocharged version of the OM 617 engine (designated OM 617 "A") for installation into their larger 300 SD line of cars. Pilot production of the engine was started in 1977, and production for the U.S. began early in 1978.

Table 3.2-1. Historical Survey of Mercedes Benz Diesel Cars (Ref. 15)

Type	Engine	Production Year	Displacement, cm^3	Stroke, mm	Bore, mm	Max. Output at Speed, (Din) kW	rpm	Torque at Speed, Nm	rpm	Number of Cylinders	Dry Engine Weight, kg
260D	OM 138	1936	2545	100	90	33	3300			4	
170D	OM 636	1949	1697	100	73.5	28	3200	96	2000	4	
180D	OM 636	1953	1767	100	75	31	3500	101	2000	4	
190D	OM 621	1958	1897	83.6	85	37	4000	108	2200	4	
200D	OM 621	1965	1988	83.6	87	40	4200	113	2400	4	
200D	OM 615	1967	1988	83.6	87	40	4200	113	2400	4	
220D	OM 615	1967	2197	92.4	87	44	4200	126	2400	4	
240D	OM 616	1973	2404	92.4	91	48	4200	137	2400	4	
300D	OM 617	1974	3005	92.4	91	59	4000	172	2400	5	
200D	OM 615	1976									195
220D	OM 615	1976		----------	As Above	----------					197
240D	OM 616	1976									197
300D	OM 617	1976									229
300CD	OM 617	1977									229
300SD	OM 617A	1978	2998	92.4	90.0	85	4200			5 Turbo-Charged	244

Table 3.2-2. Mechanical Data of Latest Mercedes Benz Four- and Five-Cylinder Naturally Aspirated Automotive Diesel Engines (Ref. 16)

Data		OM 616		OM 617	
Bore mm	(in.)	91.	(3.58)	91.	(3.58)
Stroke mm	(in.)	92.4	(3.64)	92.4	(3.64)
Bore/Stroke Ratio		0.98		0.98	
Displacement cm^3	(in^3)	2404.	(147)	3005.	(183)
Number of Cylinders		4		5	
Maximum Output (SAE net hp)		62		77	
Rated Speed	(rpm)	4200		4000	
Weight	(kg)	203		234	

Compared to the naturally aspirated (NA) five-cylinder OM 617 diesel, the turbocharged version displays an increase in performance on the order of 43 percent, with a weight increase of only 7%. In conjunction with their newly developed 300 SD sedan, the turbocharged OM 617 "A" shown in Figure 3.2-1 is the latest step in the evolution of MB-developed diesel cars.

A published paper (Ref. 15), describes the OM 617 engine as follows: The crankshaft is forged and is supported by six tri-metal bearings. The torsion damper is integral with the front pulley, which also drives the accessories. The overhead camshaft is chain driven and features a hydraulic chain tensioner and guide (Figure 3.2-2) which was found necessary in order to cope with torque fluctuations induced by the injection pump. The connecting rods are forged and annealed. The pistons are of cast high silicon aluminum alloy and have three rings. All of these oscillating parts are identical with the four-cylinder models. The cylinder block and head are of cast iron alloy. No valve seat inserts are required. Cylinder head design is essentially the same as that of the four-cylinder engine, but drilled cooling passages are provided in order to improve cooling of the bridge areas between the inlet and the outlet valves.

The exhaust manifold is a dual flow design which minimizes flow losses and exhaust pressure differences between the individual cylinders. This, in conjunction with long inlet pipes that have a pronounced ram effect, plus a

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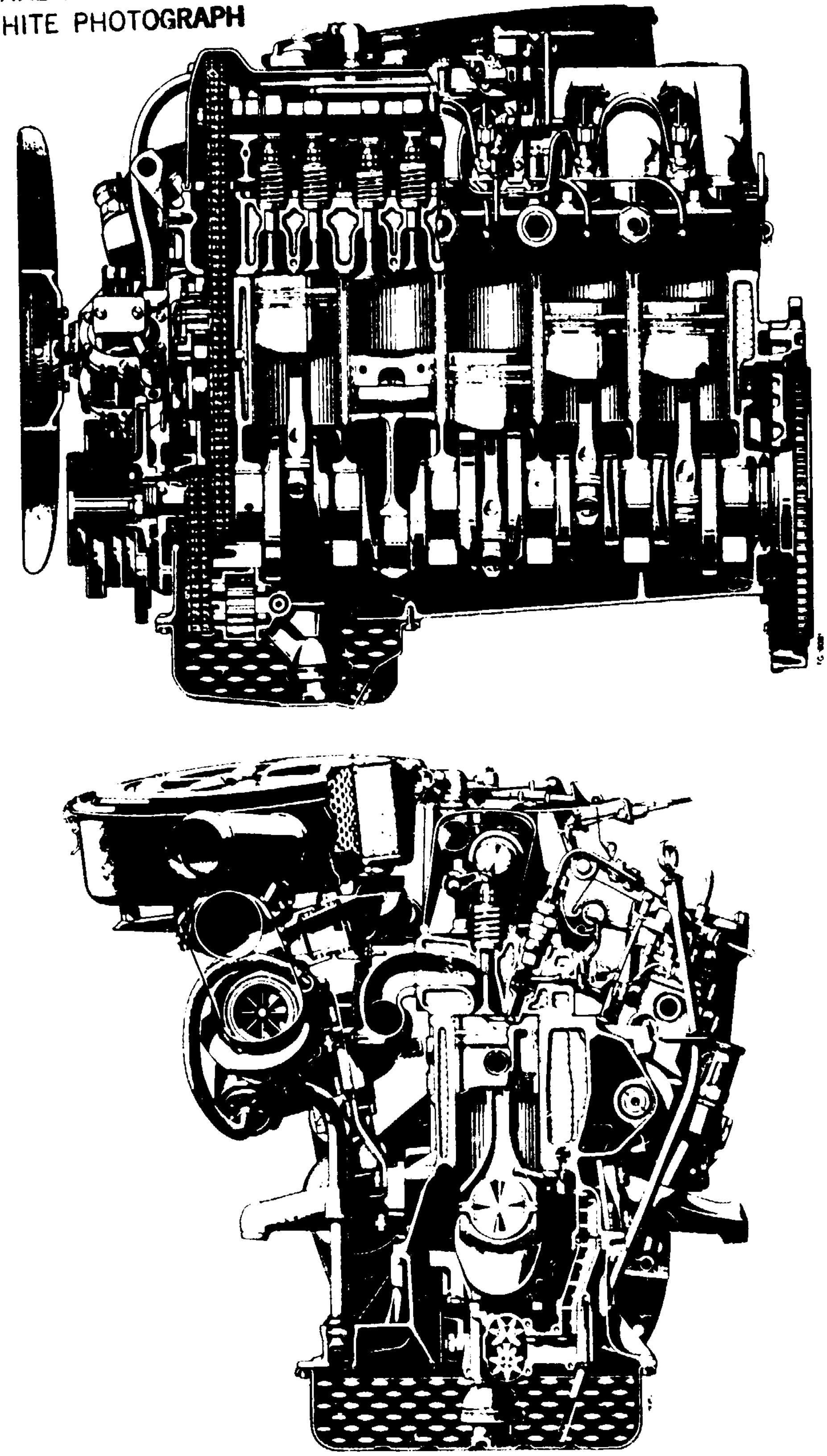

Figure 3.2-1. Longitudinal and Cross-Sectional View of Mercedes Benz OM 617A Automotive Diesel Engine (Ref. 15)

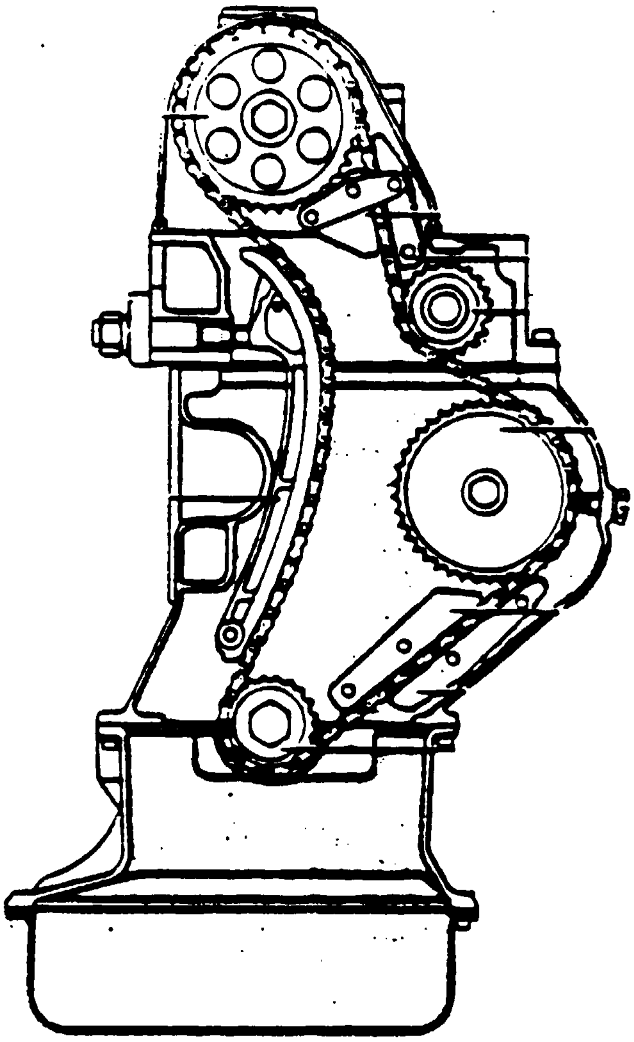

Figure 3.2-2. OM 617 Chain Drive (Ref. 16)

small valve timing overlap, has proven very effective in achieving a uniform volumetric efficiency between cylinders. Because of the strong influence of oil temperature on engine wear, the OM 617 engine has been equipped with a thermostatically controlled air/oil heat exchanger, which assures a short warm-up and also meets the high demands for external cooling during operation at high thermal loads.

The OM 617 engine uses a Bosch 5-plunger in-line injector pump which meters and controls the injection volume in the usual manner by an upper and lower helix lip in the rack-and-pinion driven plungers. The time duration of each injection depends on the ambient pressure and the engine load in such a way as to obtain the optimum fuel economy, performance, emissions, and noise levels. Injection timing is controlled by means of a centrifugally operated timer.

In contrast to all four-cylinder MB engines which have a pneumatic governor, the OM 617 five-cylinder is equipped with a mechanical governor which matches the delivered fuel to the air flow in the idle and maximum speed ranges. Between these ranges, the driver controls the fuel flow with the gas pedal to produce the required torque. According to MB, the use of a mechanical governor has a number of advantages. Rod motion control is more uniform (which helps to minimize exhaust emissions) and maximum engine speed is more precisely controlled. An inlet throttle, originally required to generate the control vacuum for the pneumatic governor of the four-cylinder engines, is no longer needed.

The engine uses pintle nozzles that operate at a relatively low injection pressure on the order of 120 bar (approximately 1800 psi). In order to reduce the initial heat release gradient and engine noise, the nozzles are tuned so that fuel delivery during the ignition delay period is as low as possible. The injection pressure lines are equipped with reverse flow dampening valves which allow for free flow in the direction of the nozzle while return pressure waves, which are mainly responsibile for after-injection, are effectively dampened. The quantities of fuel introduced due to afterinjection are negligibly small, but they do noticeably increase HC and carbon monoxide (CO) emissions, because they are introduced too late to permit complete combustion. Although described by MB as well proven and reliable, nozzles of this type have reportedly shown a tendency to foul up easily and to develop a "rattling" noise.

The supercharged version of the OM 617 engine (OM 617A) uses a turbocharger developed by Garrett/AiResearch with a pressure-activated waste gate that limits the absolute boost pressure to a maximum of 1.7 bar (Figure 3.2-3). The relatively flat operating characteristics of the vaneless centrifugal compressor permits a favorable match so that at engine speeds above 1600 rpm, compressor efficiency could be maintained between 65 and 74% (Figure 3.2-4). The turbine is sized to provide for maximum performance at medium engine speeds and loads. The waste gate bypass opens as necessary to avoid overpressurization and excessive back pressure rise because of increased pumping losses at higher engine speeds. With the bypass system, the turbocharger has been shown to be much more responsive during transient operations than a fixed geometry, uncontrolled turbocharger.

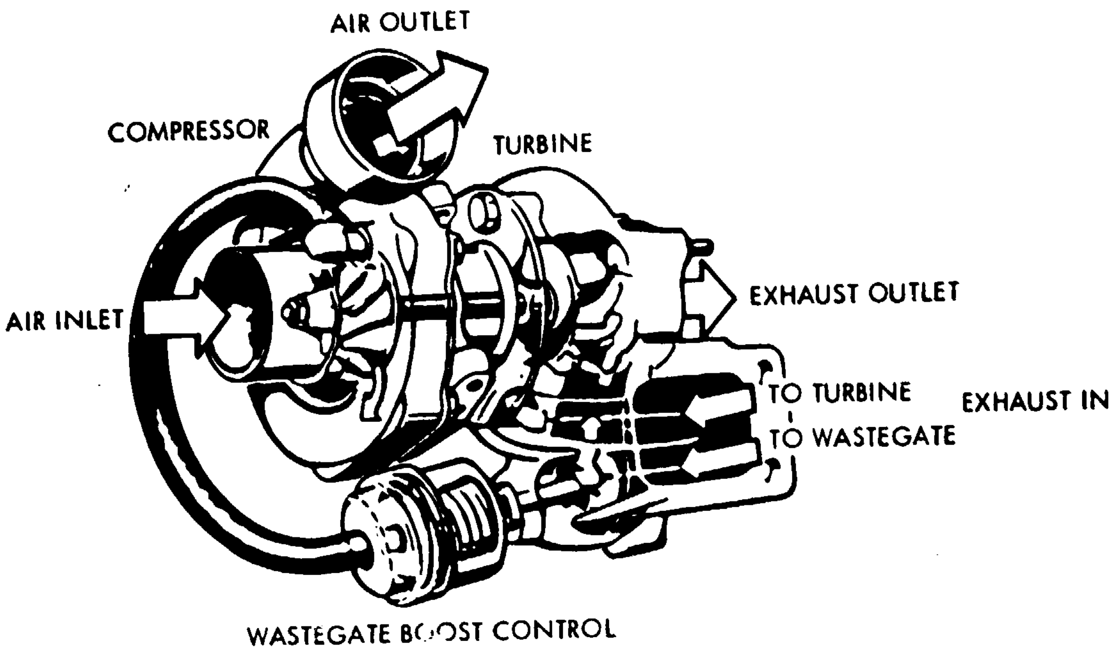

Figure 3.2-3. Garrett Turbocharger with Integral Waste Gate Control (Ref. 15)

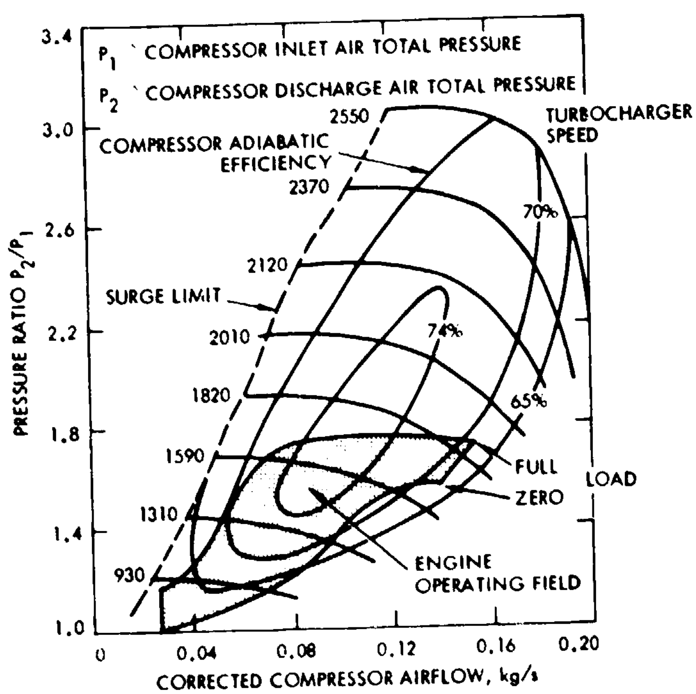

Figure 3.2-4. Engine Operating Field Shown Within Compressor Characteristics (Ref. 15)

The OM 617A engine is of essentially the same design as the OM 617 (Table 3.2-2). However, a number of changes and new design features not readily apparent were introduced to cope with the higher mechanical and thermal loads, which are about 50% higher than those encountered in the naturally aspirated OM 617 engine. The crankshaft was bath-nitrided to effect an increase in hardness and fatigue strength. The main bearings are the same as those in the OM 617 engine except for the axial thrust bearing which displayed fatigue cracks in the thrust collars and had to be changed from an integral design, to a design with separate thrust rings. The connecting rod bearings of the OM 617 engine exhibited severe damage under turbocharged operation and extensive testing was required to design a new bearing capable of coping with the higher loads, while retaining the same dimensions.

The pistons had to be redesigned and were equipped with cooling oil jets which squirted oil into an integrated ring channel underneath the piston crown. This reduced the maximum piston temperature to an acceptable level, on the order of 275° C. The oil was fed and metered to the ring channels by means of calibrated jets which aim from a stationary manifold at the open ends of the ring channel in the lower end of the piston skirts. From there, the oil is transported up to the ring channel by the reciprocating motion of the piston. Piston noise generation was minimized by narrowing the piston clearance in the lower part of the skirt, and by increasing the wrist pin bearing diameter. The axial length of the pistons was extended and the connecting rods were shortened correspondingly.

The valves were modified to cope with the higher combustion pressure by providing more material around the heads for improved rigidity. The exhaust valves are sodium cooled to make the stems more heat resistant and warp-proof. To provide the oil for the above described piston cooling system, a new, enlarged, chain driven oil pump was incorporated. This required design changes in the crankcase area.

The cylinder head gasket is a critical element in all diesel engines, particularly in those which are supercharged. The pre-chamber concept is of great advantage here because the gasket is not in direct contact with the high temperatures. However, a new gasket had to be developed and the cooling around the most critical area between adjacent cylinders had to be greatly increased. The new gasket corresponds in form with that of the naturally aspirated engine but a newly developed elastic inner material is used that cures during engine operation and, improves the long-term compression characteristics of the gasket. As mentioned earlier, cooling slots were used between cylinders to improve cooling on MB's naturally aspirated four and five-cylinder engines since 1974. On the supercharged engine, the cooling slots between the cylinders are open toward the gasket to maintain a low gasket temperature between cylinders that assures reliable sealing of gas and water.

Highest thermal loads are encountered at the bottom of the pre-chamber and in the throat between the pre- and main chambers. A good conductive path that dissipates excess heat into the cylinder head is of utmost importance. To accomplish this, the contact area between the pre-chamber and

cylinder head was enlarged. Combustion optimization tests also led to a change in pre-chamber volume and to a different arrangement of the fuel orifices for the supercharged engine.

The injection system for the supercharged engine is manufactured by Bosch and is basically the same as the one used on the naturally aspirated engine, with schedule changes that allow for the increased fuel demands of the turbocharged engine. The fuel quantity is increased as the boost pressure rises to a certain maximum flow rate in a manner that produces the same fuel-to-air ratios and exhaust smoke characteristics as the ones obtained in the naturally aspirated engine. From this point on, the fuel quantity increases with boost pressure at a lower rate. The combustion mixture becomes leaner as the boost pressure rises, which additionally reduces NO_x and smoke development.

To cope with a higher injection pressure of 140 bar, the Bosch pintle nozzle (Figure 3.2-5) was modified by introducing a longitudinal bore in the center of the pintle (Detail "A") which is connected to the plenum chambe by means of a transverse bore. This produces a more stable flow at low needle lifts in the early injection phase, and is less prone to clogging than the standard Bosch nozzle. The arrangement has also been found to almost eliminate the pinging cold start noise which is usually caused by clogging of the annular slot around the pintle.

Design changes were also necessary to accommodate the turbocharger and associated ducting. Because the Garrett turbine is of a single scroll design, the dual exhaust of the naturally aspirated engine was changed to a single exhaust leading directly into the scroll inlet. The turbocharger is rigidly flanged to the exhaust manifold. The ducts are flexible to allow for expansion and engine movement, and to reduce the transfer of acoustical noise between the components.

The performance characteristics of both versions of the OM 617 five-cylinder engine are compared in Figure 3.2-6. The turbocharged version has 46% more torque than the naturally aspirated engine at the same speed, which was obtained by precise tuning of the turbocharger and its waste gate control. With a fixed geometry turbocharger, the mid-range torque would be lower, and maximum torque would occur at high speeds where it is not needed.

Figure 3.2-7 shows the relationship between engine speed, and engine and supercharger performance criteria at full power operation. As mentioned earlier, a low exhaust smoke (Figure 3.2-7a) in the main driving range above 2500 rpm is achieved by a waste gate which limits the boost pressure to 1.7 bar and allows for progressively leaner combustion with increasing engine speeds from the point where the waste gate starts to open. The waste gate also insures that the maximum BMEP occurs at low engine speed. The engine inlet temperature (Figure 3.2-7d) does not exceed 100°C (212°F) which is favorable from the materials and the NO_x emission standpoint. The turbine temperature does not exceed 800°C (1470°F), a value that is very conservative for today's gas turbines.

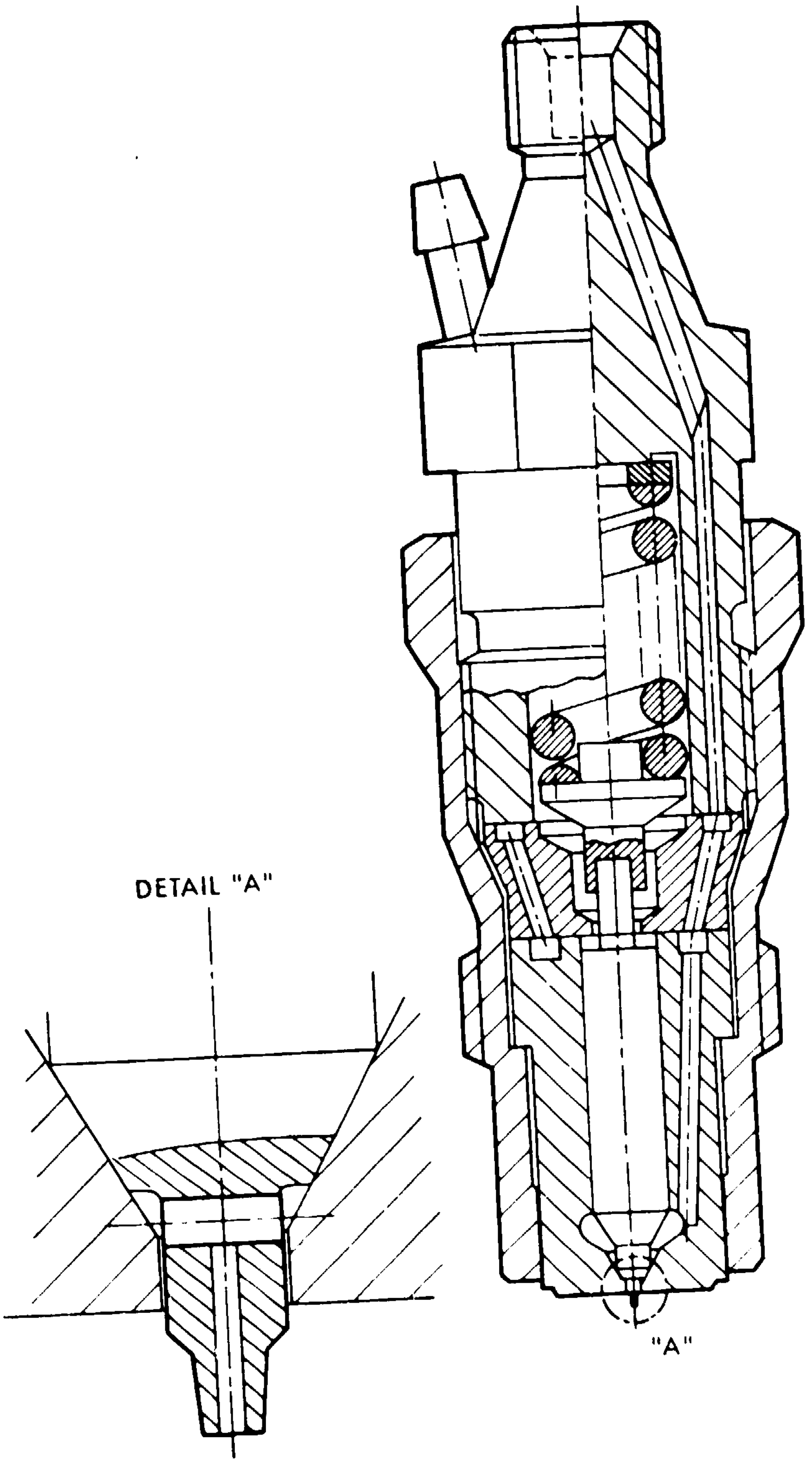

Figure 3.2-5. Bosch Injection Nozzle with Central hole in Pintle (CHIP) Nozzle (Ref. 15)

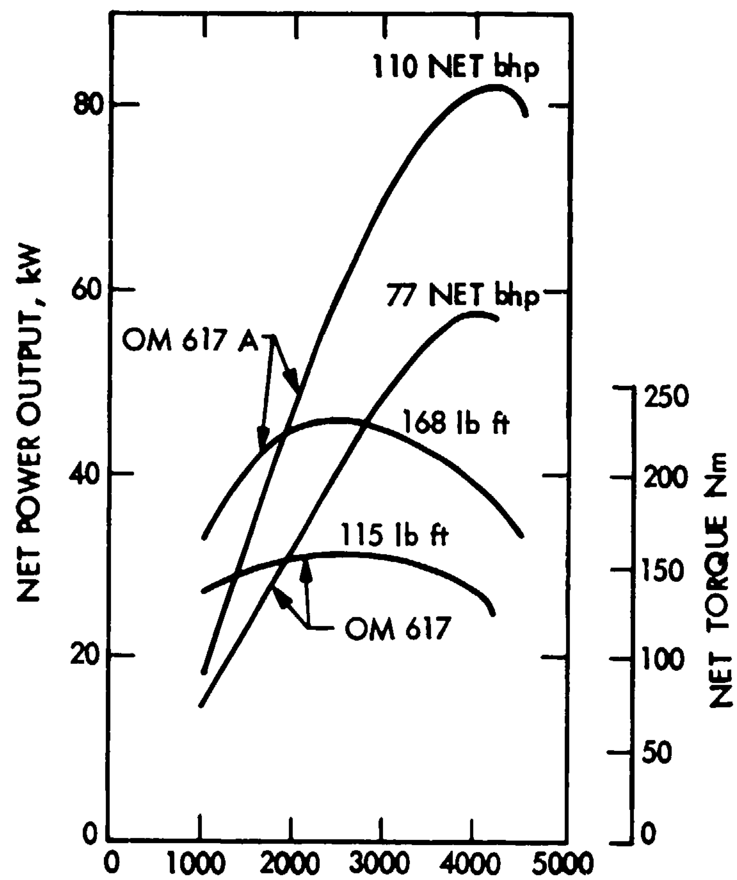

Figure 3.2-6. Comparison of Engine Performance: Naturally Aspirated OM 617 vs Turbocharged OM 617A (Ref. 15)

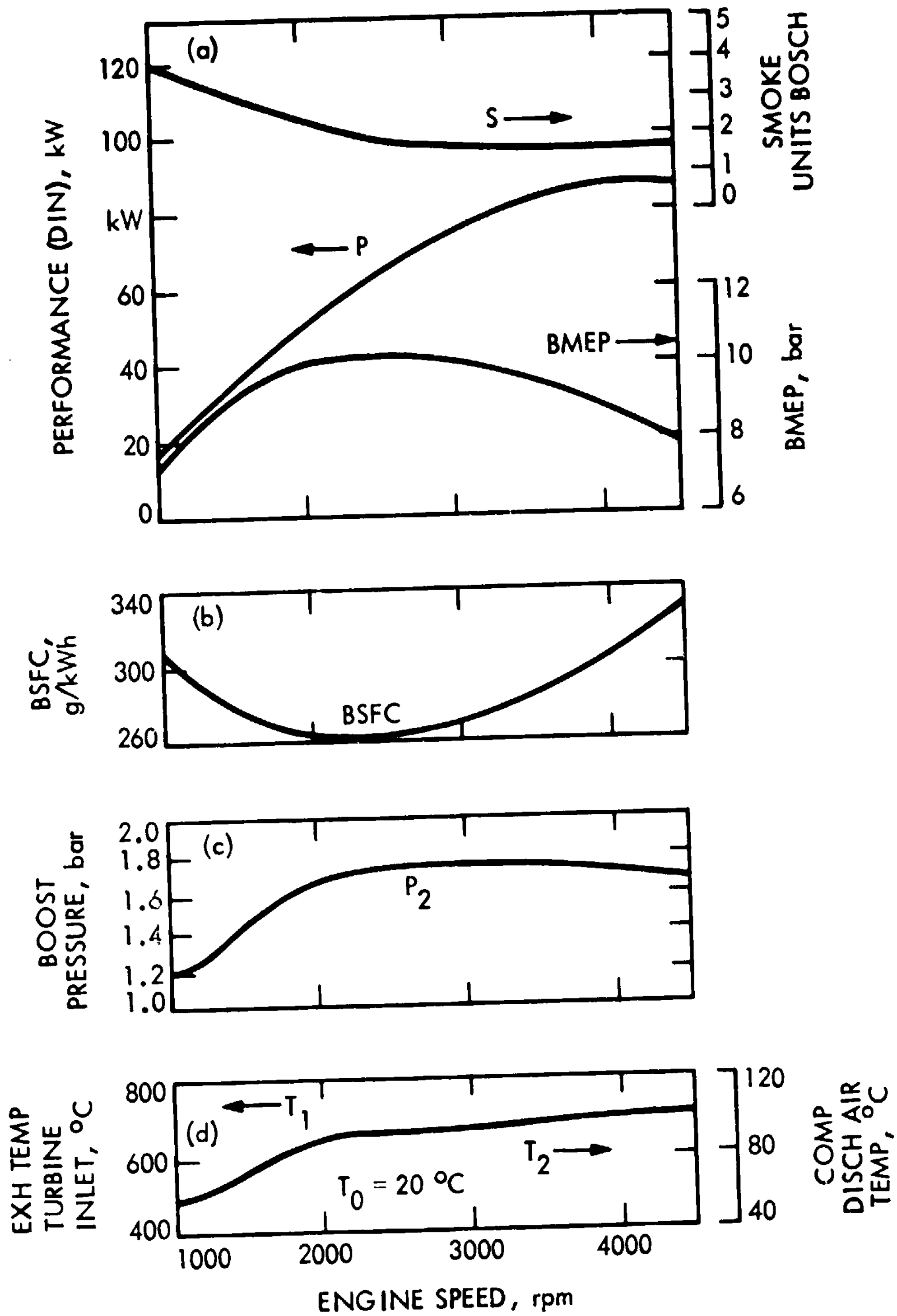

Figure 3.2-7. Engine and Turbocharger Characteristics at Full Load (Ref. 15)

The successful adaptation of the turbocharger is also apparent from the flatness of the fuel consumption map (Figure 3.2-8). The fuel consumption curves closely parallel road-load demands over a wide range of engine speeds and power settings. The optimum and near-optimum fuel consumption islands center about the part-load speed and load range most frequently used in urban driving. With 245 gal/kWh, the fuel consumption of the supercharged version is 20 gal/kWh (9%) better than that exhibited by the naturally aspirated OM 617 engine.

Mercedes Benz diesel cars currently produced have a key start system. Turning the key energizes the glow plugs and a yellow light appears on the dashboard. The time for glow plug operation is indicated by the yellow lamp and is dependent upon engine coolant temperature in the cylinder head. The glow plugs remain energized until the engine has started, but glow operation is limited to 150 s to avoid excessive battery drainage.

Figure 3.2-9 shows the progress that has been made over the years from the 180 D model (1953) to the present time with regard to acceleration. This progress has been achieved by gradually reducing engine weight, improving power per unit displacement, and by reducing the weight of the cars as well. With a 0 to 60 mph acceleration time of 13 s the 300 SD has equal or faster acceleration than most gasoline powered cars. With a 0 to 60 mph time of 21 s, the naturally aspirated 300D is considered sluggish by U.S. standards.

Figure 3.2-10 compares the road-load vehicle fuel economy of the MB 240 D, 300 D, and 300 SD. All models display superior fuel economy at low vehicle speeds. Compared to the 240 D, the fuel economy of the 300 D is slightly lower, because vehicle weight is higher and a greater number of cylinders is involved, which results in a reduced BMEP and a higher specific fuel consumption. Mercedes Benz says this effect could be minimized by reducing the rear axle ratio to about 6% slower ratio.

Figure 3.2-11 compares fuel economy and exhaust emissions of the 300 D and SD under highway, urban and composite driving conditions. A fuel economy advantage of the supercharged 300 SD over the 240 and 300 D models is evident (Figure 3.2-11B), and would be even more apparent if compared to a 300 SD car equipped with a naturally aspirated engine of equivalent power. The high torque brought about by turbocharging made it possible to operate at a lower rear axle ratio and closer to the optimum specific fuel economy island of the engine map. The turbocharged version is slightly higher in NO_x emission (Figure 3.2-11A) but is considerably lower in CO and HC, which is also indicative of a lower particulate emission. Both versions have been improved to comply with U.S. statutory emission standards for the duration of the waivers for NO_x and particulates.

Figure 3.2-12 compares the interior noise levels of the 300 D and the 300 SD diesels to the 230 series Mercedes Benz gasoline powered car for the 125 to 1000 Hz octave band spectrum at speeds of 30, 55 and 80 mph. At high speed there is no significant difference between the diesel- and the gasoline-powered cars. At all speeds, the octave band levels decrease uniformly with increasing frequency. There are no dominant peaks at any vehicle or engine speed. Despite the fact that the five-cylinder engine poses some inherent

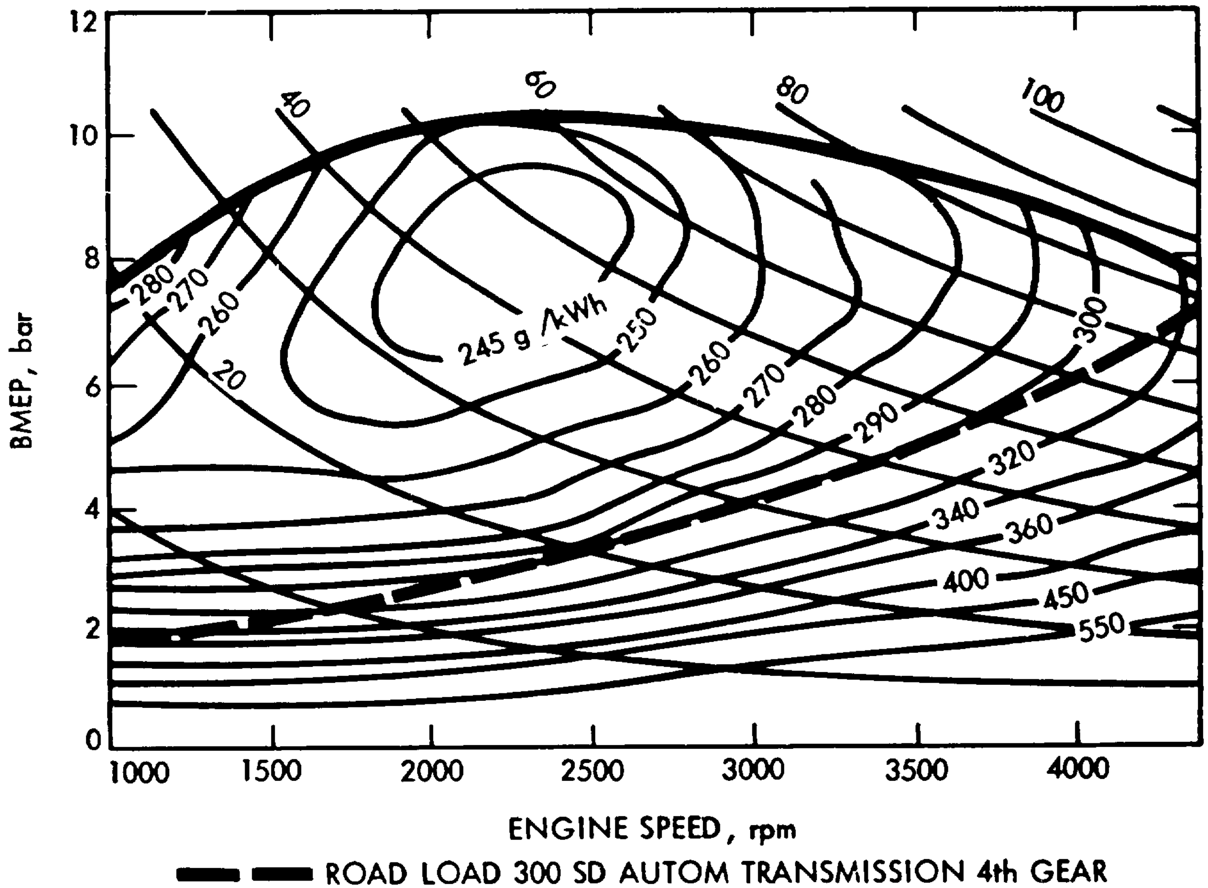

Figure 3.2-8. Fuel Consumption Map of Mercedes Benz OM 617A Five-Cylinder, 3-ℓ Turbocharged Automotive Diesel (Ref. 15)

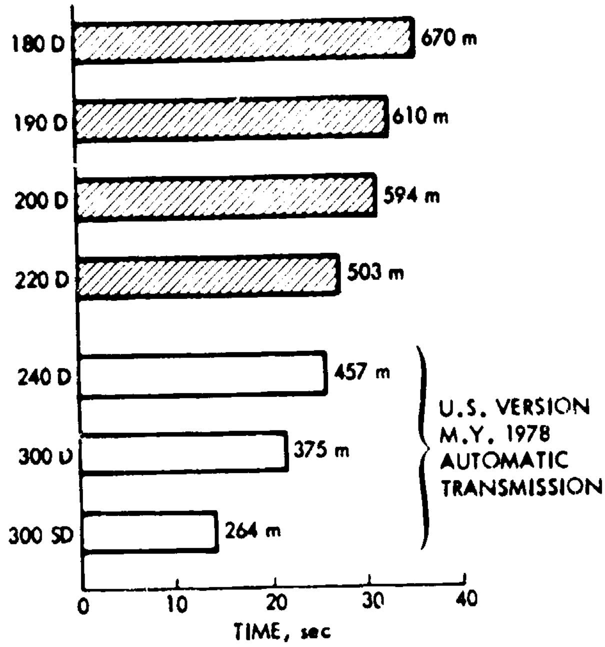

Figure 3.2-9. Historical Survey of Acceleration Times and Distances of Diesel Cars - 0 to 60 mph

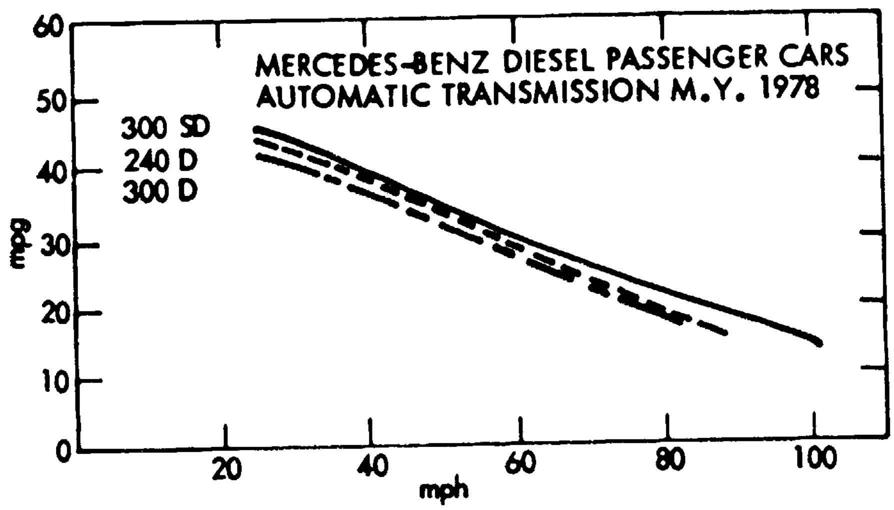

Figure 3.2-10. Comparison of Fuel Economy at Road Load (Ref. 15)

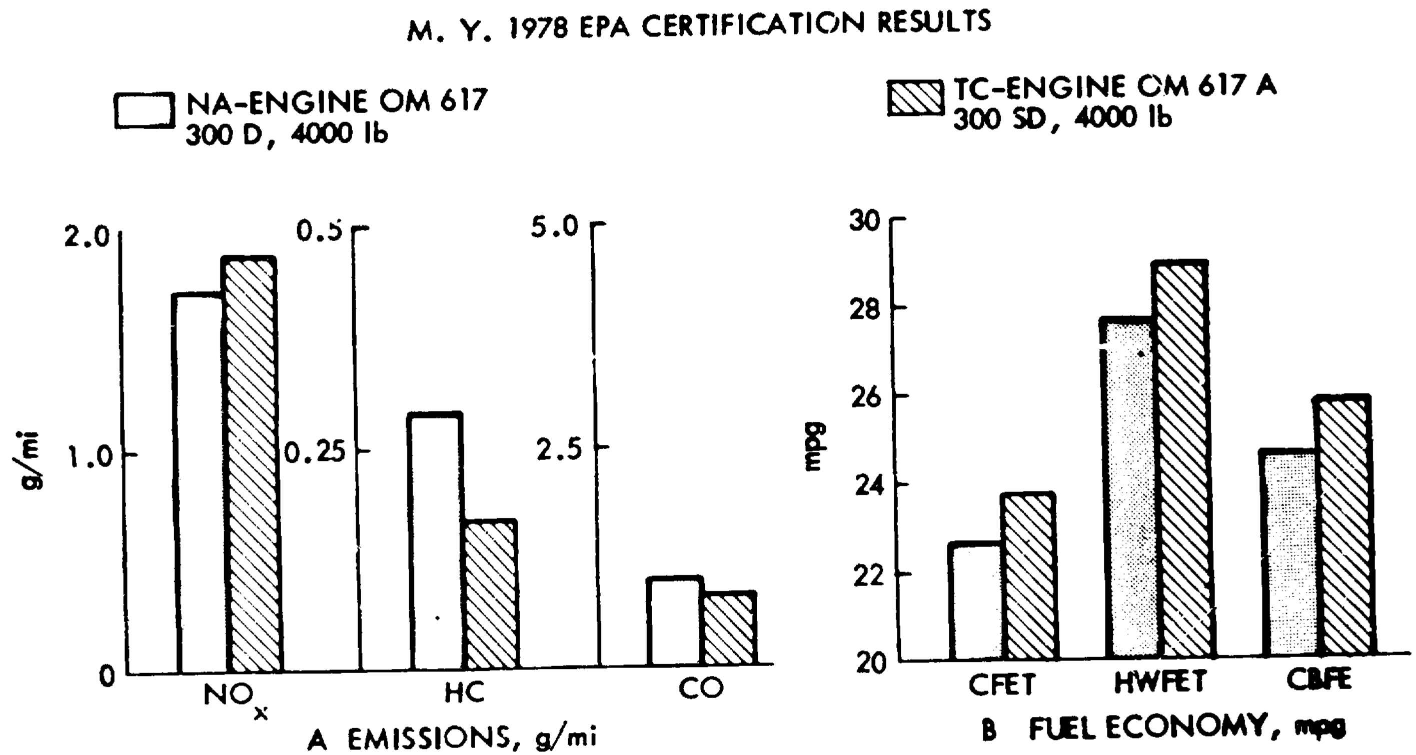

Figure 3.2-11. Comparison of Emissions and Fuel Economy of 300D with NA-OM-617 Engine versus 300SD with TC-OM-617A Engine (Ref. 15)

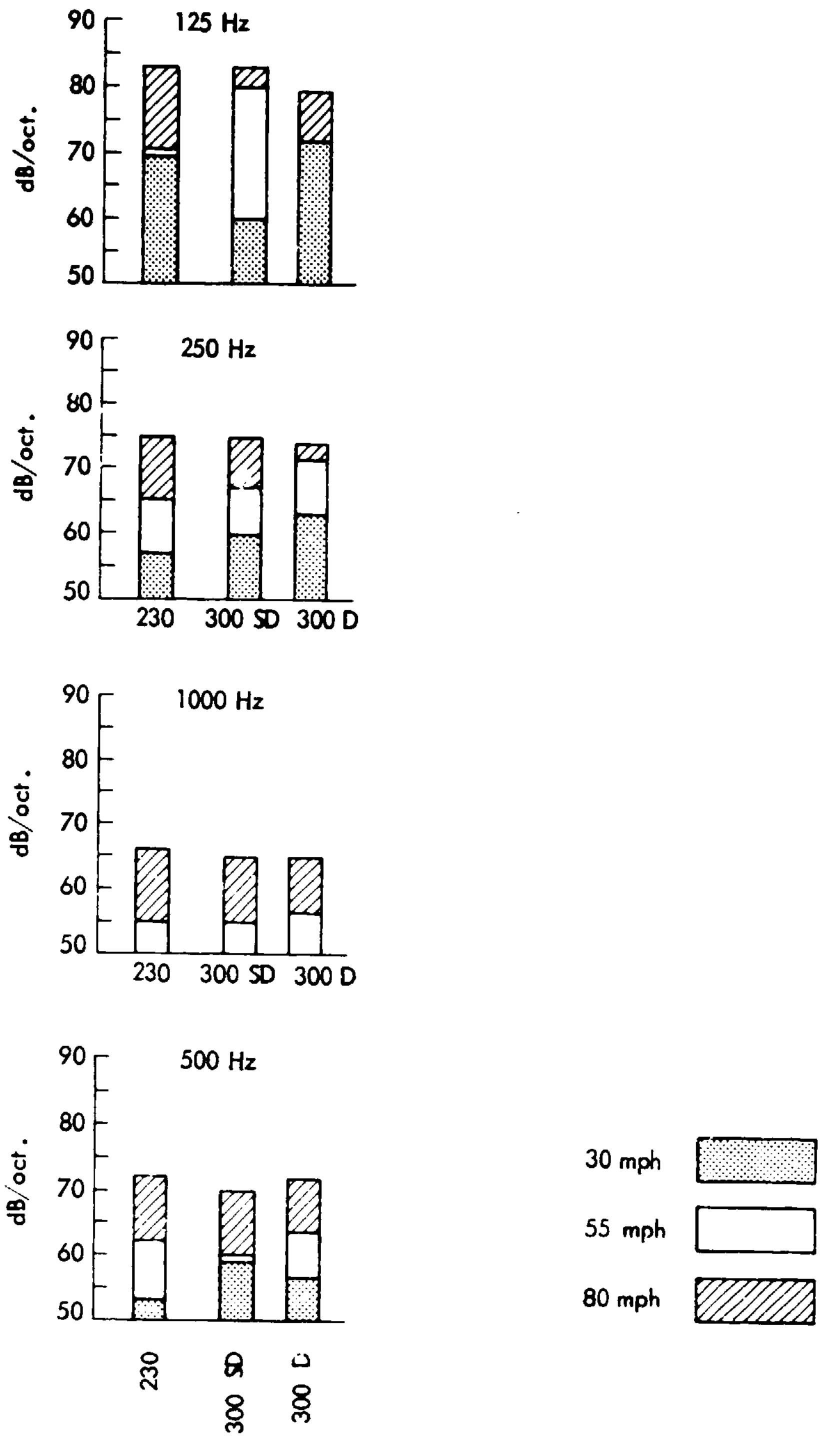

Figure 3.2-12. Effect of Speed on Interior Noise Levels (Octave Band Spectra) of Mercedes Benz Diesel Cras (Ref. 15)

dynamic problems, generally excellent noise behavior was attained by carefully tuning and dampening the engine suspension and attached components, by reducing noise radiation from the engine, and by reducing the frequency response of the car to the lowest possible level. However, MB feels that noise generated during low speed full power acceleration must be further reduced to favorably compare with the lower noise level of a gasoline powered car.

Mercedes Benz has not revealed any plans for the future at this time. Indications are that R&D work is concentrated primarily on injection refinements and the implementation of electronically controlled exhaust gas recirculation (EGR). Mercedes Benz has reportedly developed a new crankcase ventilation system (Ref. 17) that keeps the engine oil from mixing with blow-by gases, thereby reducing oil contamination from carbon particles, which becomes a severe problem with EGR. The system reportedly works so well that MB has considered raising the oil change interval from 3000 to 5000 miles on their existing engines.

Despite the uncertain future for diesel cars in the United States, MB is continuing present marketing, and forecast an increase in total sales in the U.S. Vehicles included are the 240 D, 300 D, 300 SD and a 300 CD coupe. A station wagon, designated the 300 TD (T for touring and transport), powered by an OM 617A engine, has recently been added to the line of Mercedes Benz cars. Current plans limit U.S. diesel sales to 65% of their production, presumably depending upon the extension of the emission waiver.

3.3 PEUGEOT

Peugeot has been producing diesel cars for many years, but it was not until 1974 that it introduced the XD90 diesel as an option for their 504 series sedans and station wagons in the United States. The XD90 engine was a new design and is manufactured on a separate production line.

As shown in Figures 3.3-1 and 3.3-2 the engine is a conventional four cylinder push-rod-overhead valve design which feeds into a Ricardo-type Comet V swirl chamber. The engine block is cast iron, and the heads are aluminum. The crankshaft has five bearings. A Bosch EP/VM 2200 HR 12 type injection pump is used. The Peugeot 504 diesel is 2 mm larger in bore and stroke than the 504 gasoline engine, and displaces 2.112 ℓ. It develops 62 bph at 4500 rpm, and a maximum torque of 91 ft-lb at 2000 rpm, versus 92 hp and 120 ft-lb developed by the 504 gasoline engine with a 1.971-ℓ displacement. The engine weighs 480 lb including clutch housing and gearbox, compared to 381 lb for the gasoline option. The 0 to 60 mph acceleration time obtained with 504 diesels averaged 23.6 sec, and attained a maximum speed of 84 mph, versus 16.2 sec and 98 mph with the 504 gasoline engine.

In 1977, the displacement of the 504 diesel was increased to 2.3 (Table 3.3-1), which boosted the rated power from 62 to 71 hp and reduced the 0 to 60 mph acceleration time from 23.6 to 21 sec (Ref. 19). The 504 series of diesels accounts for approximately 20% of all its passenger car deliveries. Peugeot intends to further strengthen its commitment to diesels,

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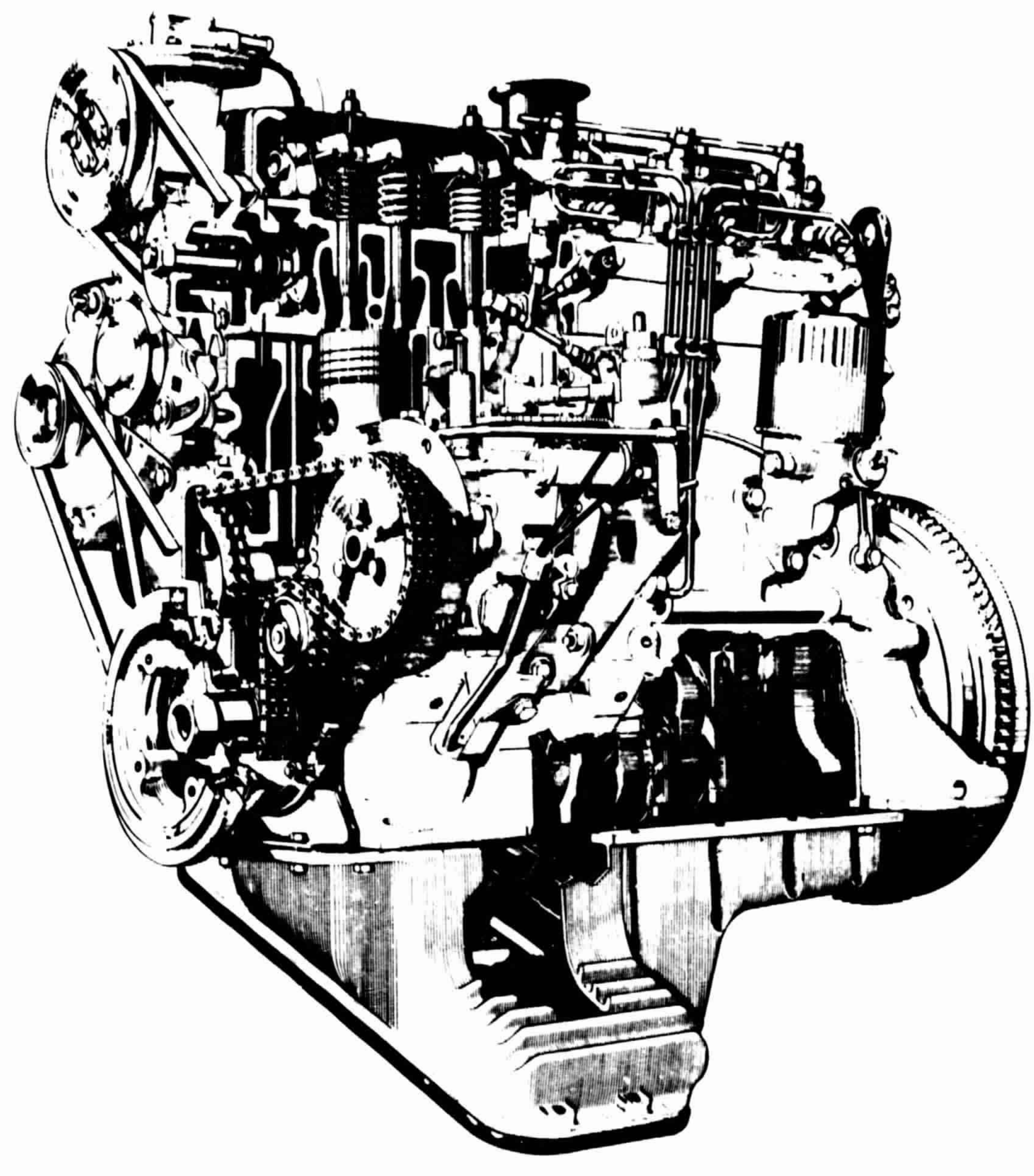


Figure 3.3-1. Peugeot 2.1-ℓ Engine for 504 Series Sedans and Station Wagons (Ref. 18)

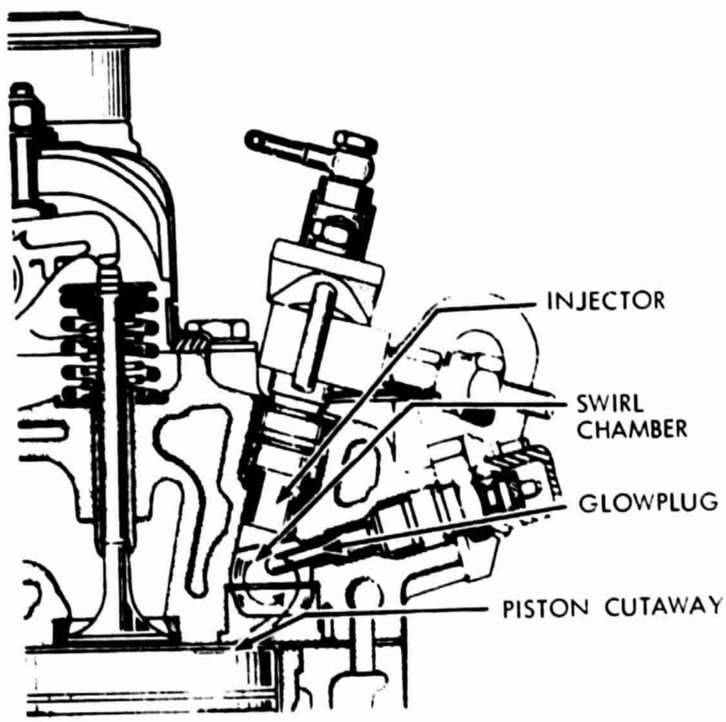


Figure 3.3-2. Peugeot Combustion Chamber Design for 2.1-ℓ 504 Diesel Engine (Ref. 18)

Table 3.3-1. Peugeot Engine Data (Ref. 18)

Type	504 NA		505, 506 Diesel	
	Gasoline	Diesel	NA	TC
Number of Cylinders	4	4	4	4
Bore (mm)	88	90		
Stroke (mm)	81	82		
Displacement (cc)	1971	2112	2300	2300
Nominal power (hp)	92	62	71	80
@ (rpm)	5600	4500	4500	4100
Maximum Torque (ft-lbs)	120	91		136.0
@ (rpm)				2000
Compression Ratio		22.2	22.2	21
Engine Weight Including Clutch & Gearbox (lb)	381	480		
0-60 mph Acceleration (sec)	16.2	23.6	21	17.5
Maximum Speed (mph)	98	84		
Boost Pressure, Bar (gauge)	NA	NA	NA	0.6

and according to press information (Ref. 20), will introduce two new models - the 305 GRD and a turbocharged version of the 2.3 diesel.

The 305 GRD four-cylinder engine, which will be marketed in the U.S. by 1983, is designed to mount transversely for front wheel drive, and with a displacement of 94.4 cid (1.547ℓ) is obviously aiming at the market now taken by the VW Rabbit diesel. It has a chain drive overhead camshaft and, as with all other Peugeot diesels, uses a Ricardo Comet V swirl chamber. Head and block are of a light alloy integral design with wet liners. The center crankshaft bearing (one out of five) ties into the block with vertical and horizontal rods for increased stiffness. Compression ratio is 22.5 and as on other Peugeot engines, the injection system is by Bosch.

A turbocharged version of the 2.3-ℓ engine (Figure 3.3-3) will be introduced in 1981 in the U.S. by Peugeot to power their line of 505 cars. Turbocharging will boost engine power from 71 to 80 hp at a lower rpm (4100), with a flatter torque characteristic that produces 136 ft-lb at only 2000 rpm. No details of specific internal design changes are available at this point in time, except that the engine has been reinforced to take the higher loads associated with turbocharging. The compression ratio has been reduced from 22.2 to 21. The turbocharger is a Garrett/AiResearch T03, which is currently used in gasoline-powered Ford Mustangs and GM pickups. A spring-loaded waste gate limits the maximum boost pressure to .6 bar. Equipped with a 5 speed transmission the TC 505 has demonstrated a 28/36 mpg EPA rating and a 17.5 sec, 0-60 mph acceleration capability, compared to 29/35 mpg and 23.5 sec for the naturally aspirated version. The price the consumer pays for improved driveability is $1000, or approximately 10% over the naturally aspirated version.

Peugeot is one of the smallest diesel producers. Reportedly, Peugeot was the first diesel manufacturer capable of having their naturally aspirated 2.3-ℓ 504 model car certified to a 1.0 g/mi NO_x standard in 1980 (Ref. 17). This car will also meet 1981 California standards, but the turbocharged

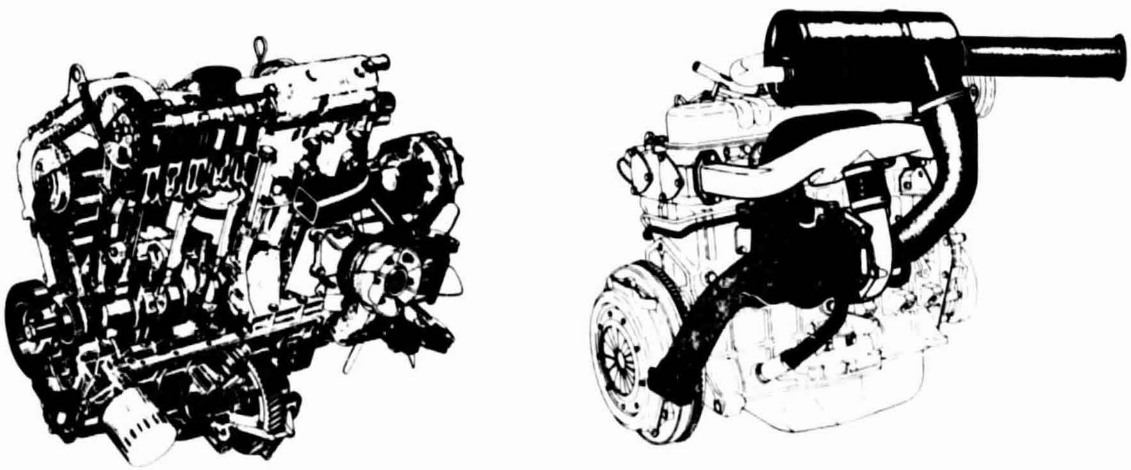


Figure 3.3-3. Peugeot 604 Turbocharged 73-hp Engine (Ref. 21)

version will not. As of December 1980 the turbocharged TC 505 has completed U.S. certification with the 1.5 NO_x waiver. Peugeot R&D work is concentrated primarily on developing a modulating EGR system and on improving the fuel injection system.

3.4. VOLKSWAGEN

Volkswagen is a newcomer in the automotive diesel field, but has taken the lead in the sub-compact and compact class with both the naturally aspirated and turbocharged engines. Volkswagen can be considered most competent and successful where the diesel conversion of existing gasoline engines is concerned. Fortunately, VW has published and presented a wealth of information (Refs. 22, 23, 24, 25) regarding design detail and performance of their diesel engines, as well as information pertaining to design rationale and development approaches.

The VW 1.5-ℓ, 50-hp diesel is a comparatively recent development. It was initially introduced in Europe as a power plant for the VW Golf car, and was marketed in the U.S. beiginning in 1976 as an option for the car which is now designated "Rabbit" here. The engine has evolved from the 1.6-ℓ VW gasoline engine which has been marketed in the U.S. since 1974. Reportedly, the engine was originally designed with a later dieselization in mind. As shown in Figures 3.4-1 and 3.4-2, it is a four-stroke, four-cylinder overhead cam design using a swirl chamber combustion system generally patterned after the Ricardo design, with combustion pressure loading only slightly higher than in the gasoline version. The engine is fitted with a Bosch Model VE distributor type injection pump with a mechanical governor which is similar to the one used by Mercedes Benz. The cylinder head is cast aluminum and the main cylinder block is of cast iron. Although both are closely based upon existing VW gasoline engine design, both have proven very adaptable to diesel use with regard to stiffness and temperature. There have been no problems in sealing the cylinder head to the block in the diesel version. No changes have been necessary in bearing design and materials, and crankshaft design is adequate for handling the diesel combustion loads. The pistons are aluminum alloy of a three-ring design with a "Ni-resist" insert in the top ring groove, and have a special crown design that integrates most efficiently with the pre-chamber combustion system. The engine has a bore of 76.5 mm and an 80-mm stroke. At 286 lb, it weighs only 37 lb more than a 1.1-ℓ equivalent performance gasoline engine.

A unique feature is a special Pirelli-designed nylon and rubber cogged belt system which was developed after extensive testing. As can can be seen from Figure 3.4-1, the cogged side of the belt drives the injection pump and the overhead cam. The smooth side of the belt drives engine accessories which do not require fixed gearing. A tuned intake manifold reduces the gas-flow losses across the engine, and as shown in Figure 3.4-3, achieves volumetric efficiencies of up to 90% that are relatively high for a small high speed diesel engine. According to VW, production engines of the described design will meet the performance criteria shown in Figure 3.4-4 within the indicated tolerances (Ref. 23). The rated output of 50 hp is obtained at the 5000-rpm maximum speed. The 1.6-ℓ gasoline engine for the VW Rabbit is rated at 70 hp at 5000 rpm.

A comparison of the road load fuel consumption of the gasoline and diesel versions of the Rabbit engine is shown in Figure 3.4-5. It is seen that at

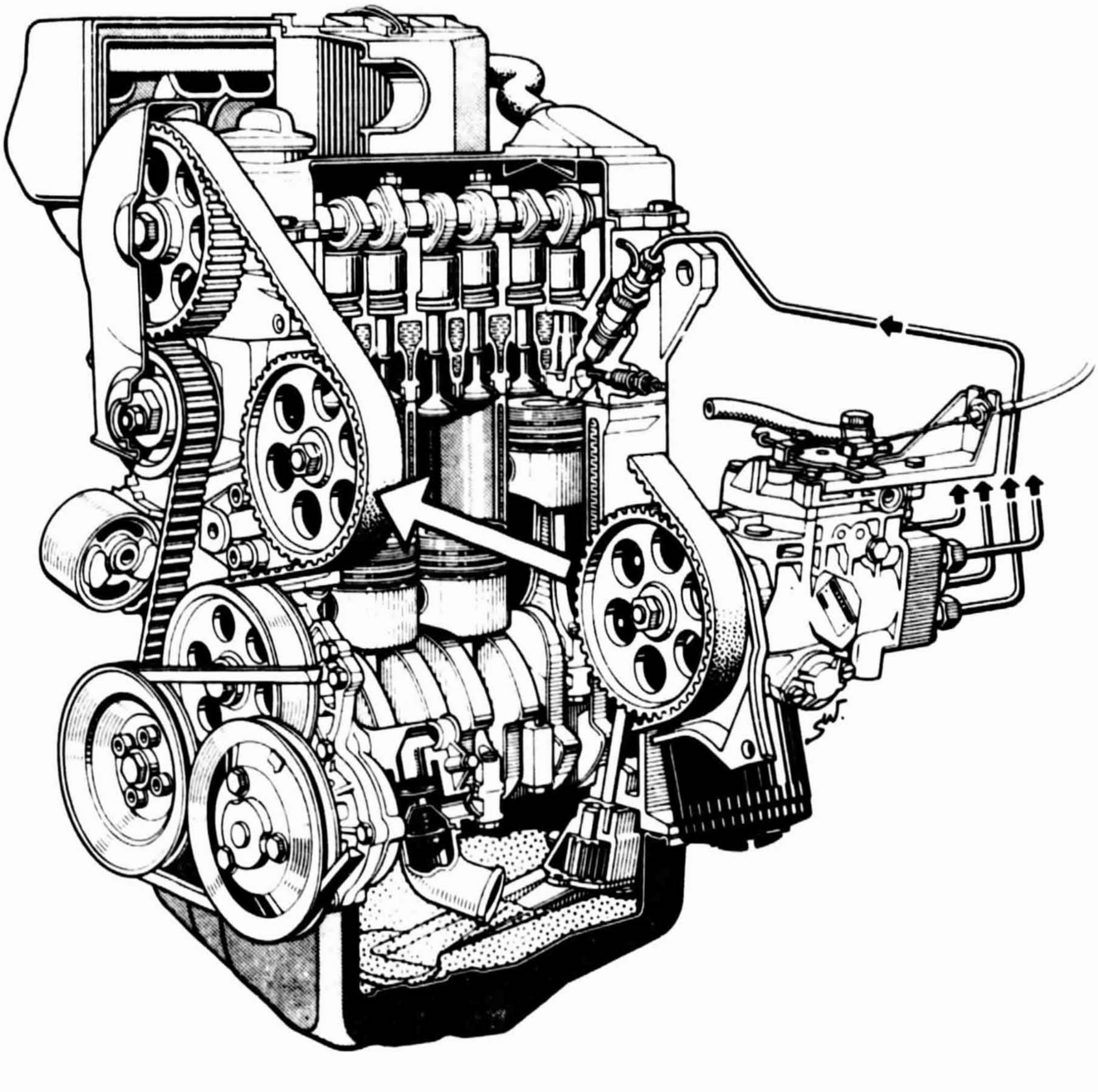


Figure 3.4-1. Cut-Away View of 1.5-ℓ, 50-hp, VW Automotive Diesel Engine (Ref. 25)

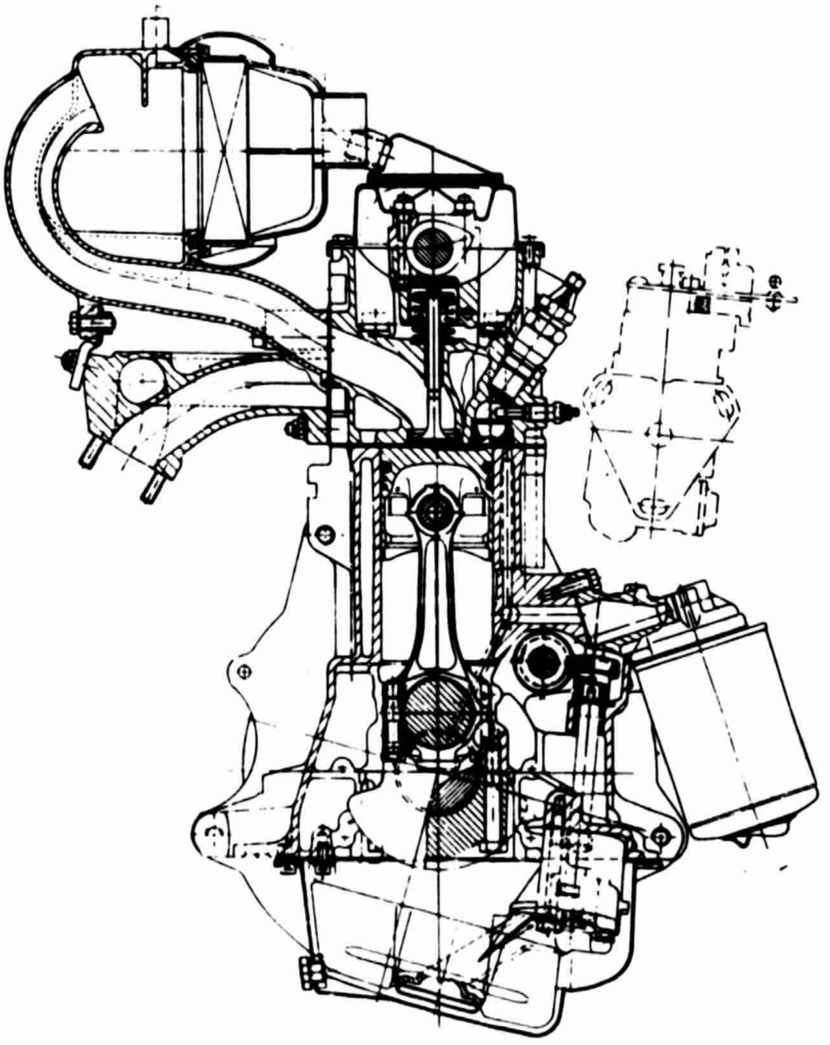


Figure 3.4-2. Cross-Sectional View of 1.5-ℓ, 50-hp, VW Automotive Diesel (Ref. 25)

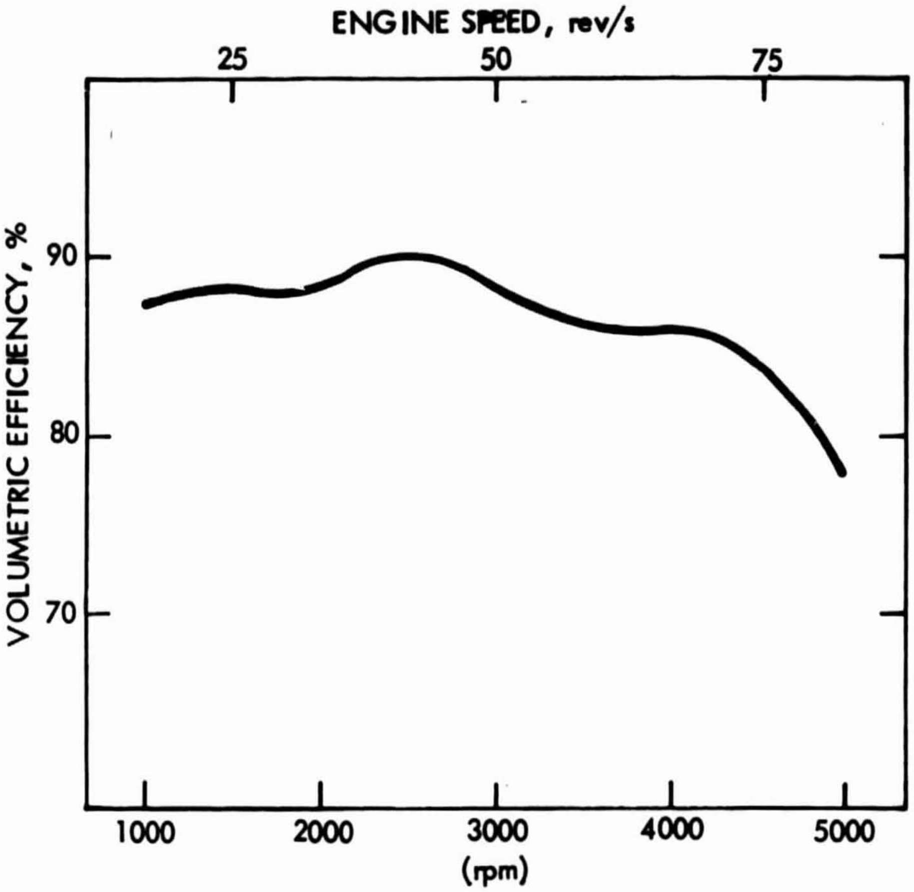


Figure 3.4-3. Effect of Engine Speed on Volumetric Efficiency, 1.47-ℓ Swirl Chamber Engine (Ref. 23)

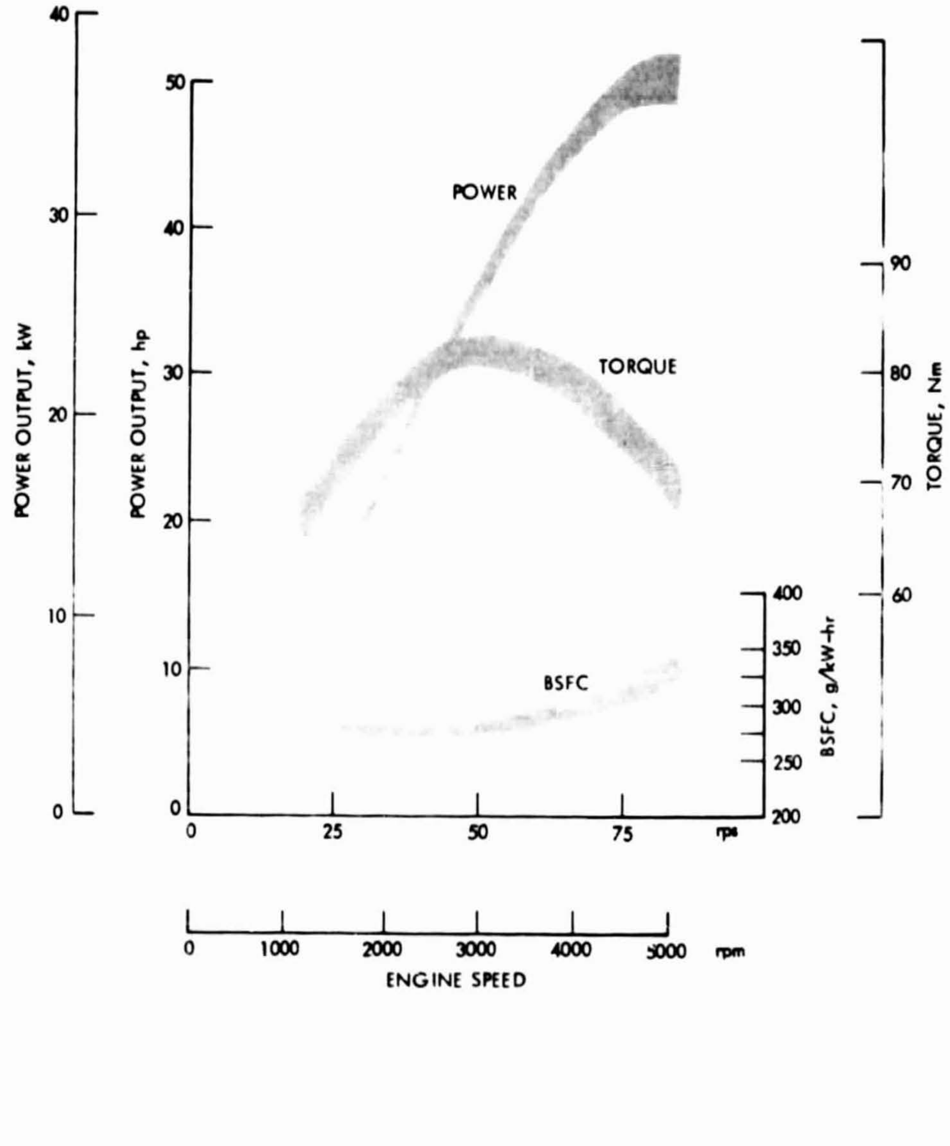


Figure 3.4-4. Performance Scatter Bands of VW 1.5-ℓ , 50-hp Diesel Engine (Ref. 23)

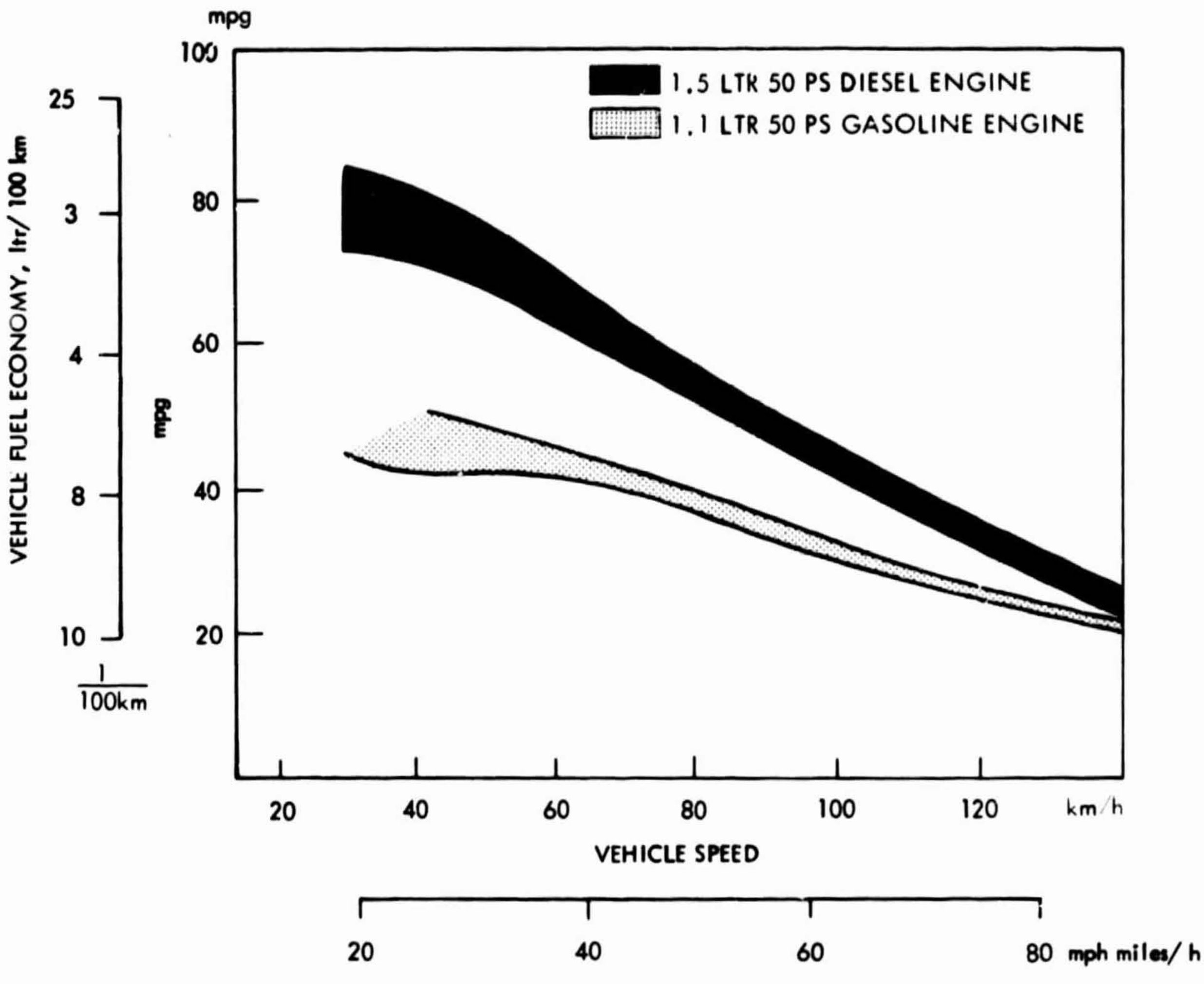


Figure 3.4-5. Comparison of Road-Load Fuel Economy for VW Rabbit: 50-hp Diesel versus Equivalent Gasoline Engine (Ref. 24)

low speed the gas mileage of the diesel is substantially higher (up to 100 percent) than that of the gasoline engine equivalent. For composite driving, VW claims that their diesel car gives superior fuel economy over the gasoline version of equal displacement on the order of 25 to 40%.

The Environmental Protection Agency currently rates the Rabbit diesel first in fuel economy, according to the latest 1981 data for 49 states (Ref. 26). The Rabbit diesel car is rated 42/56 mpg with a four-speed transmission, 38/56 mpg with a five-speed transmission, and slightly less for the California versions. The production diesel-powered Rabbit meets current emission standards for CO, HC, and NO_x, and will just meet 1981 emissions standards, including the one for particulates. The car cannot meet 1980 California or 1983 federal standards without the 1.5-NO_x waiver.

Tests with a turbocharged five-speed version of the diesel Rabbit car have been in progress since 1978. The turbocharged engine resembles the 1.5-ℓ Rabbit diesel with slight modifications of the combustion chamber and hardware changes as necessary to cope with higher operating pressures, temperatures, and stresses. A turbocharger developed by Garrett/AiResearch is used. Figure 3.4-6 shows the performance data of the turbocharged 1.5-ℓ engine in comparison with a naturally aspirated gasoline engine having the same rated power output. The engine develops 70 hp at 5000 rpm, which is 20 hp more than the naturally aspirated Rabbit diesel engine. According to VW, the top speed demonstrated with a 2250 lb VW test car was approximately 100 mph, and the car achieved 80 mpg at 30 mph, 62 mpg at 50 mph, and 21 mpg at top speed. Up to 55 mpg was obtainable as an average in city driving and 69 mpg on highways. The 0 to 60 mph acceleration time demonstrated was on the order of 13.5 sec, which is better than the gasoline powered Rabbit car. Compared to other diesels (see Figures 3.7-1 and 3.7-2), the fuel economy gains obtained with the research, turbocharged Rabbits are extraordinarily high, which suggests that fuel cut-off during deceleration and coasting was used in test runs.

For the present time, VW relies on expanding the existing four-cylinder naturally aspirated diesel into a line of five- and six-cylinder engines. A five-cylinder version became available as an option for the Audi 5000 in mid-1979. It develops 67 hp at 4800 rpm, which is relatively low power for a car that weighs 3000 lb. The 0 to 60 acceleration time of the diesel powered Audi 5000 is on the order of 17 to 19 s. This is rather sluggish by U.S. standards. A turbocharged version of the five-cylinder Audi will reportedly be available for the 1982 model year (Ref. 26).

The six-cylinder version is earmarked for use in light commercial vehicles. Retaining the same cylinder dimensions, the engine has a displacement of 2.383-ℓ, and produces 75 (DIN) bhp at 4500 rpm. This engine will temporarily be adopted by Volvo, and be fitted into the Volvo 242 coupes, sedans, and 245 wagons. Volvo is developing a six-cylinder diesel of its own, but adopting the VW engine will lower capital expenditures and permit quicker entry into the diesel market in the 1980s.

Although the existing 1.5-ℓ Rabbit diesel meets the 1.0 g/mi NO_x standard by a narrow margin, VW had joined GM's request for a 4-year 1.5 g/mi NO_x waiver in order to introduce a slightly larger (1.6-ℓ) four-cylinder diesel for the Rabbit and Dasher models to improve air-conditioned performance.

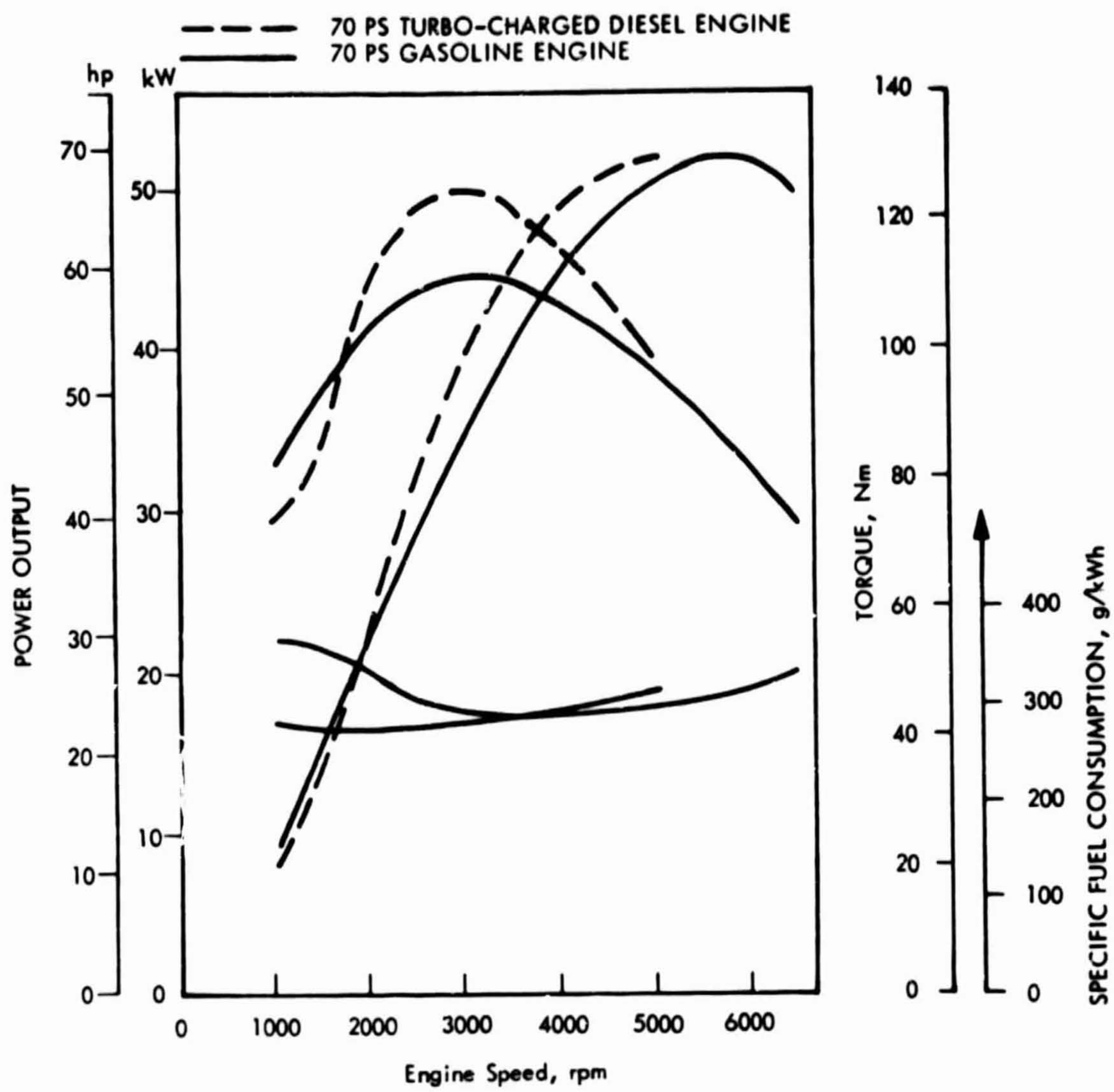


Figure 3.4-6. Comparison of Performance Data: Turbocharged Rabbit Diesel versus Gasoline Engine of Equivalent Performance (Ref. 24)

The 1.6-ℓ diesel has a new crankshaft with a larger stroke, a new block, new pistons, and more oil capacity.

The rationale for development in the direction of a modular unit production system is the result of trade-off studies conducted at Volkswagen which have shown that cylinder units between 300 and 400 cc produce the highest specific output in terms of power-per-unit swept volume. According to a published paper (Ref. 23), for cylinder sizes smaller than 300 cc, the surface-to-volume ratio of the combustion chamber increases, which leads to increased heat losses and lower combustion efficiency. The loss in specific output associated with smaller cylinder size cannot be compensated by raising the nominal engine speed without causing a drop in volumetric and thermal efficiency, and a steep increase in friction losses and fuel consumption. With smaller cylinder bores, it also becomes more difficult and costly to comply with the diesel tolerance requirements. Cylinder sizes of more than 400 cc (assuming certain restrictions with regard to bore-stroke ratios, piston velocity, friction losses and mass forces) require a reduction in nominal speed more than necessary to maintain a high specific output on the order of 25 kW/ℓ.

Figures 3.4-7 through 3.4-9 show the effect of the number of cylinders on the relationship between engine geometry and performance output, assuming that: (1) the single cylinder volume will remain within the bracketed values discussed above (less than 400, and greater than 300 cc), (2) the piston speed will not exceed 13.5 m/sec, (3) engine nominal speeds will not exceed 5000 rpm, and (4) the bore/stroke ratios will remain within the VW-preferred value of approximately 1.1. Within these constraints, a production line of four-, five-, six-, and a paper eight-cylinder engine cover a total displacement range of 1.2 to 3.2 ℓ . Engine weight, including accessories, radiator, water and oil is primarily a function of power, and increases with power output (Figure 3.4-7). The number of cylinders also has a strong effect. The engine specific weight shows a distinct optimum for power output and number of cylinders. (Figure 3.4-8). Minimum practical wall thickness as determined by foundry technology, as well as accessory weight (which changes only to a small extent with engine weight) are primarily resonsible for this. In any case, the use of a turbocharger makes a distinct decrease in engine weight for a given power output possible (dashed line, Figure 3.4-8). The relationship between engine displacement and box-volume is almost linear for a given number of cylinders (Figure 3.4-9). From the box-volume or packaging standpoint, four-cylinder engines are most advantageous up to 2.6-ℓ displacement. Beyond that, a larger number of cylinders leads to a more compact design. A V-configuration is of increasing advantage with larger displacements.

3.5 OLDSMOBILE

At the present time, all of the large U.S. auto makers are involved in automotive diesel engine projects, but GM is the only U.S. firm that currently manufactures passenger car diesels, of Oldsmobile design. The Olds diesel has been marketed since 1978 in the Cutlass Supreme, 88 and 98 Oldsmobile line of cars, in the Cadillac Eldorado and Seville models, and in

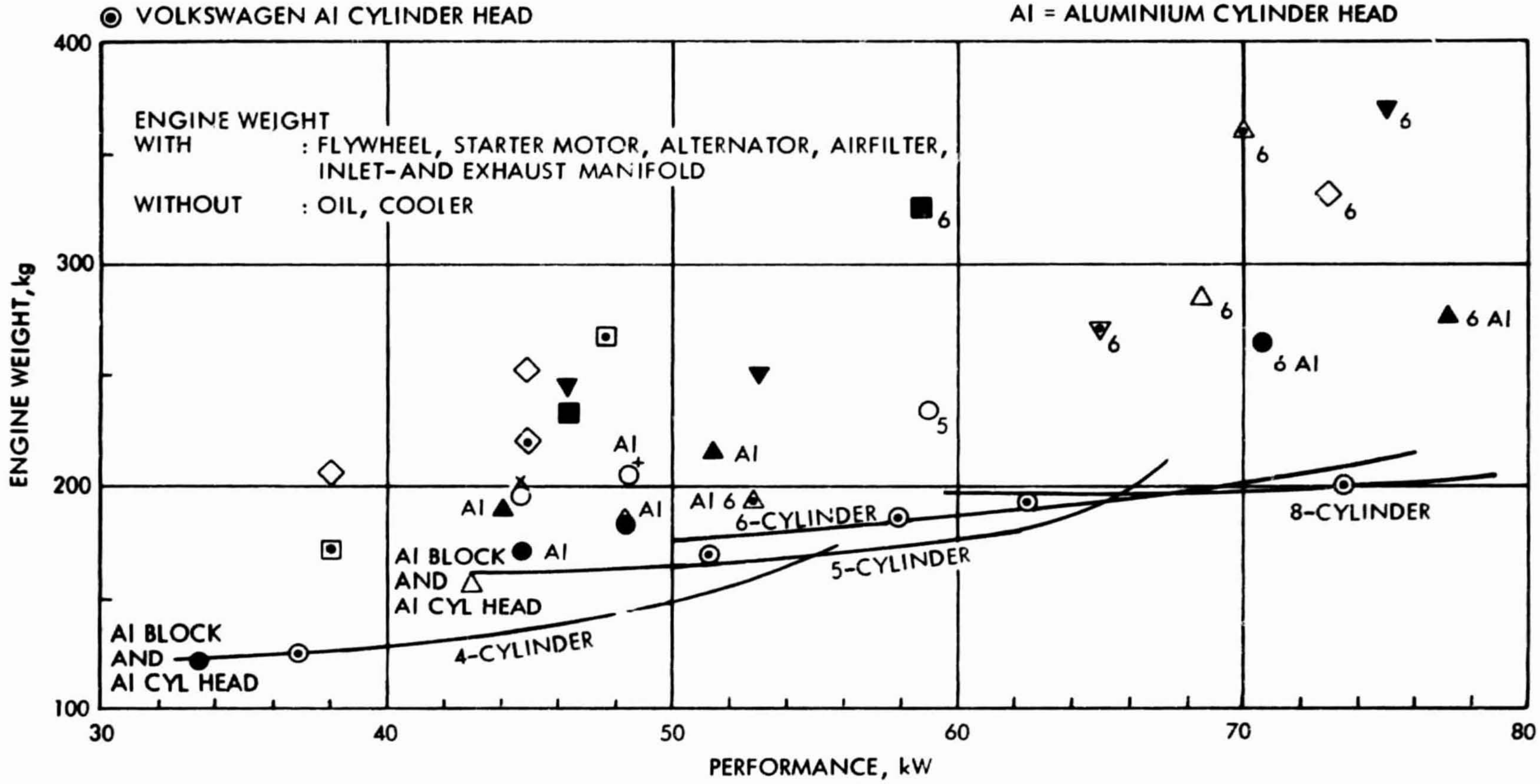


Figure 3.4-7. Effect of Number of Cylinders on Engine Weight (Ref. 25)

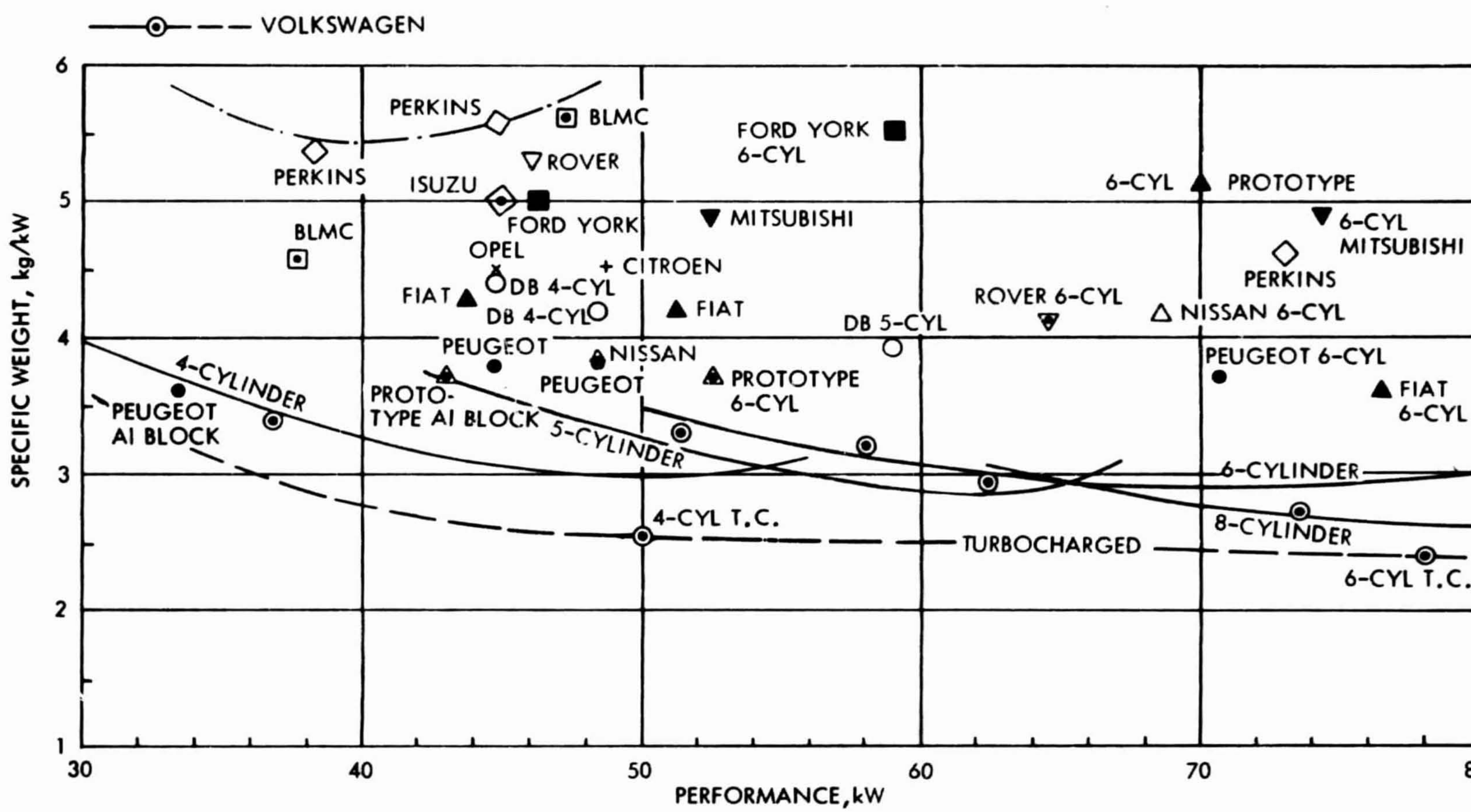


Figure 3.4-8. Effect of Number of Cylinders on Specific Engine Weight (Ref. 25)

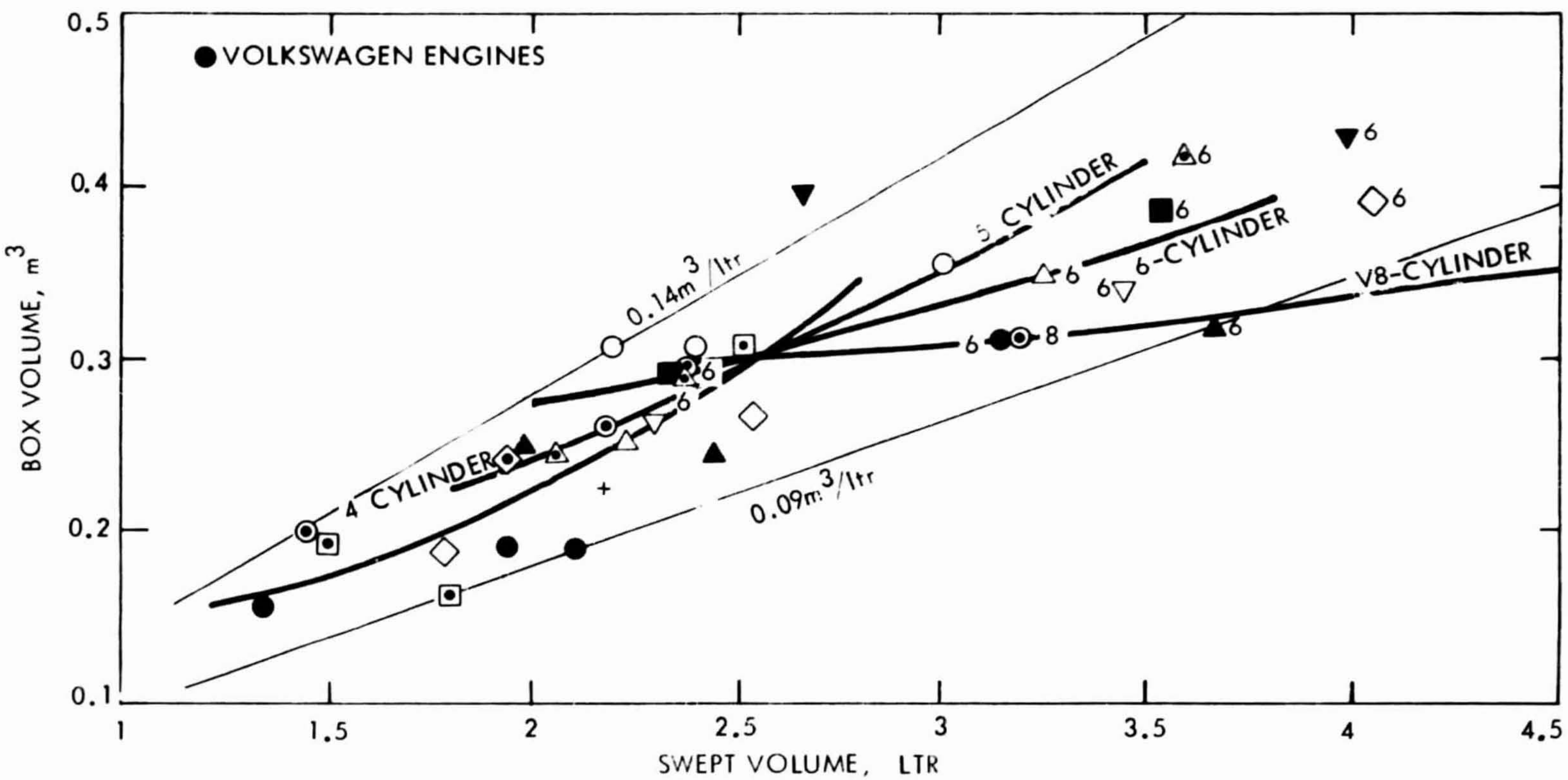


Figure 3.4-9. Effect of Number of Cylinders on Box Volume of Automotive Diesels (Ref. 25)

GM light duty pickup trucks and vans. There were 177,000 engines built for the 1979 model year (Ref. 27).

The automotive diesel development at Olds started in 1973 with an alternate engine evaluation, which included Opel and Nissan engines, and a GM V-6 engine with both direct and pre-chamber injection. This was followed by an extensive combustion chamber evaluation program that compared nearly 300 combinations of various pre-chamber and injection nozzle designs. As a result of these studies, Oldsmobile decided to proceed with the development of a 350 CID (5.7-ℓ) V-8 diesel (Figure 3.5-1) that would use as much manufacturing tooling as possible from the Olds family of gasoline engines currently in production. Oldsmobile took an approach similar to VW by converting an existing gasoline engine into a diesel rather than developing a totally new diesel engine.

As can be seen from Table 3.5-1, the major dimensions of the diesel are the same as those of the 350 CID Olds gasoline engine. However, a number of changes were introduced to achieve sufficiently clean diesel combustion, and to compensate for the increased structural loads, vibration, noise and wear. Referring to Figure 3.5-2, the cast iron cylinder head is modified to recieve the injection nozzles, the glow plug and a cast stainless steel pre-chamber, and to provide for a compression ratio of 22.5. The pre-chamber is pressed into the head. As a result of the above mentioned combustion chamber evaluation tests, a swirl chamber was chosen which has a tangentially offset throat and an inboard injection nozzle. According to Olds, this arrangement has proven favorable with regard to HC and NO_x emissions, and in the reduction of smoke and noise (Ref. 28). An interesting feature is the so-called flame slot which directs the hot gases coming from the pre-chamber into the valve pockets and at a flat angle over the piston tops. This proved to be very helpful in reducing piston heating.

The head gasket of the Olds diesel was derived from a new head gasket developed by Fel-Pro, which was introduced on Olds gasoline engines in 1977. The gasket consists of a metal core and a thin sheet of "beater add" asbestos that is chemically laminated to each side of the core with a thermoset adhesive, and is then coated with blue Teflon. In addition, the new Fel-Pro gasket features beaded silicone sealing material which is printed on both sides of the gasket to seal the cooling passages across the head-to-block interfaces. The gasket for the diesel has the same bolt pattern and basic materials as that used in the gasoline engine, but a number of extra features were added to cope with the higher temperature and combustion pressure of the diesel.

The combustion pressure of the Olds engine is on the order of 1100 psi versus 850 psi in the gasoline engine. This is relatively low when compared to 1800 psi, which is typical in GM direct injection truck diesels. To assure effective sealing of the combustion area, a stainless steel wire O-ring was incorporated that provides for sealing of extremely high pressures when clamped between the block and the cylinder head. The wire gasket runs around the main combustion chamber and has an extended tab that acts as a heat shield beneath the pre-chamber, where temperatures are highest.

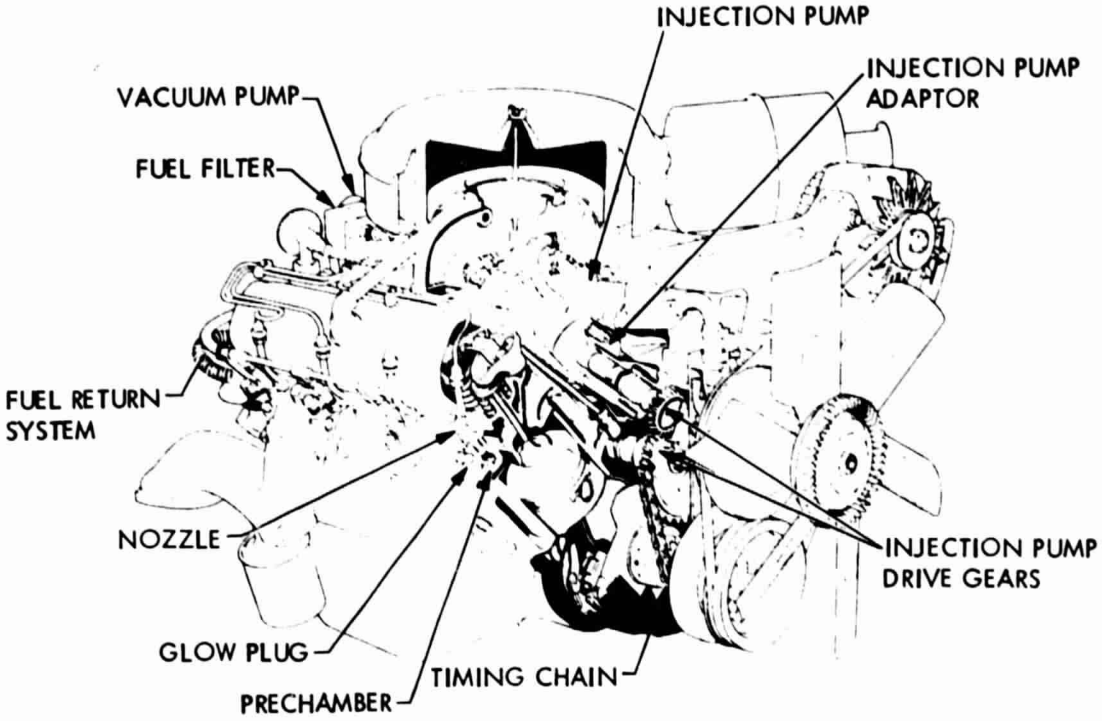


Figure 3.5-1. Cutaway View - Complete Olds V-8 Diesel Engine (Ref. 28)

Table 3.5-1. Basic Dimensions - Gasoline Versus Diesel (Ref. 28)

5.7 Liter Diesel vs. 5.7 Gasoline Engines		
	Gasoline	Diesel
Bore x Stroke	4.047 x 3.385	4.047 x 3.385
Bore Center Distance	4.625	4.625
Deck Height	9.330	9.330
Rod Bearing Diameter	2.125	2.125
Main Bearing Diameter	2.500	3.000
Rod Center to Center	6.000	5.8855
Piston Pin Diameter	.9805	1.095
Piston Pin Wall	.192	.214
Piston Weight	641 g	796 g

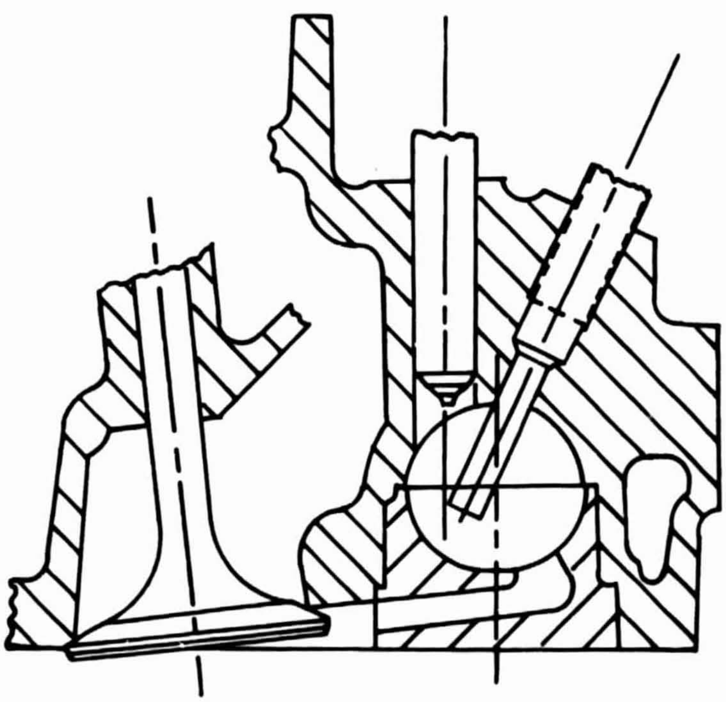


Figure 3.5-2. Section View - Olds Combustion Chamber (Ref. 28)

The intake valves were changed from 1041-1047 steel to 21-2 steel to increase high temperature strength. In the gasoline version, this material is used only for the outlet valve. To cope with increased stresses throughout the valve drive train and with impaired oil quality because of carbon contamination, the diesel valves were provided with welded-on hardened tips. The camshaft material was changed from a moderate carbide level to high carbide hardenable cast iron (Conkerall), and the lifter material was changed from sintered iron (typical of gasoline engines) to a tungsten titanium chilled iron alloy.

The pistons had to undergo significant redesign. The composition of aluminum alloy used in the gasoline engine was changed to increase high temperature strength. The transition from the skirt into the head was reinforced for improved thermal and impact resistance. Pin offset was eliminated to reduce skirt loading. The top compression ring was provided with an insert molded cast iron full groove protector, and the top ring side clearance was increased to promote the necessary free ring movement to avoid ring sticking at high temperatures. The piston pin diameter was increased from .978 to 1.11, and the pin wall section from 0.217 to 0.273 in. The connecting rods were made of an enlarged Isection for increased column load and greater bending strength. The pin base was also enlarged and was fitted with a bushing to provide for a free floating piston pin. The crankshaft structure was significantly reinforced and the main bearing diameter was increased from 2.5 to 3 in. No changes in material, heat treatment or production methods are reported.

The injection pump used by Olds is a Roosa Master Model DB2 pump supplied by the Hartford Division of Stanadyne, Inc. In contrast to the multi-plunger, in-line type Bosch pump, which has been adopted by most other diesel producers, the Roosa Master pump is an opposed rotary plunger distributor type of pump which has been modified to meet the fuel demands of the Olds engine. The pump has a centrifugal governor that controls idle and maximum speed, and an automatically controlled injection device that advances the cam timing ring of the pump as necessary to implement an optimized injection timing schedule. The nozzles are of the multihole fixed orifice type and are also supplied by Stanadyne.

The Stanadyne injection pump has been subject to a variety of consumer complaints. During the first ten-thousand miles of service life, the pump developed sealing problems around the governor weight container ring leading to rough idling, frequent stalling and complete failure. These conditions may have contributed to or have been the cause of other complaints such as excessive oiling. Reportedly all these deficiencies have been remedied, and cannot be considered typical for gasoline-diesel conversions. General Motors has introduced a redesigned pump that is warranted against failure for five years or 50,000 mi, and is retroactive on all Olds diesels produced since 1977 and used on GM cars, most of which had less the 50,000 mi at this point in time.

Figure 3.5-3 shows the typical performance characteristics of the engine. The maximum horsepower is 120 at 3600 rpm, maximum torque (220 ft-lb) occurs at 1600 rpm. The EPA rated vehicle fuel efficiencies achieved with General Motors cars and light trucks are shown in Table 2.2-1. The Olds production

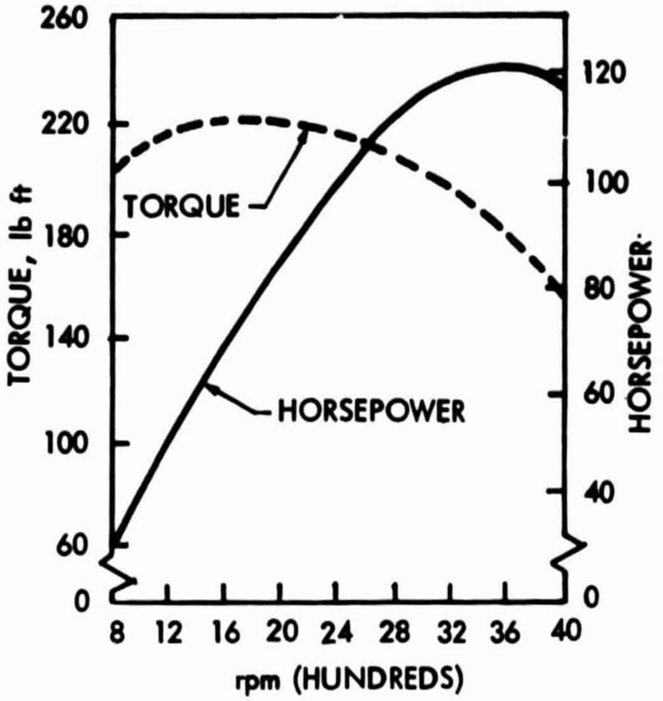


Figure 3.5-3. Performance Characteristics of Oldsmobile 350 CID Diesel Engine (Ref. 28)

line engine meet current emission standards with a comfortable margin for the duration of the 1.5 g/mi NO_x waiver. The emission deterioration behavior experienced with this engine after 50,000 mi is considered excellent. Compliance with future emission standards represents an engineering challenge for an engine of this size, and will be discussed in more detail in Section 4.0. The engine has been certified for 1.5 g/mi NO_x, including ARB 100,000 mile warranty requirements.

According to Olds 88 (4000 lb IW) and 98 (4500 lb IW) model cars powered with the above described diesel engine exhibited a performance level that is competitive with many gasoline-powered, family-size automobiles (Ref. 28). The 0 to 60 acceleration time is on the order of 16 sec. Diesel combustion noise and vibration have been kept to a minimum by a variety of means: (1) by design measures taken in the combustion, injection nozzle and timing areas; (2) by carefully tuning engine accessory mounts, and inlet and exhaust ducts; and (3) by using acoustic insulation for a high density hood blanket and in-dash insulation barrier.

Reportedly, there is no perceptible combustion noise at highway cruising speeds. There is discernible but not objectionable noise during idle and in urban traffic operations. Olds claims that smoke and odor generated by their diesels are below objectionable levels under all operating conditions.

In the Olds diesel development program, great emphasis was placed on improving the cold start capability, which is an area of traditional concern with all diesels. The Olds engine is said to start with No. 2 diesel fuel after a 12-hour soak at 10° F with no more than four seconds of cranking. As with other diesels, the starting procedure is "turn-key" with a lamp indicating that the engine is ready for starting. With a new glow plug system that was introduced in 1979, the glow time could be reduced from 60 to 6 sec (Ref. 29). The new system uses a temperature sensor in the cylinder head near the pre-chamber, and a voltage regulator which begins energizing the glow plugs at high current, then tapers off as the sensor temperature rises. The glow plugs stay on for about 60 sec after start to eliminate white smoke and to assure a smooth idle and good initial acceleration. Using a block heater (which is supplied with all Olds diesels), pre-heated batteries and No. 1 diesel fuel, starts have been demonstrated with engine ambient temperatures as low as -40°F.

3.6 BAVARIAN MOTOR WORKS

According to earlier press information (Ref. 30) BMW has been developing a 2.4-ℓ diesel engine since 1978 with plans to market the engine with 528i-series cars in the near future. Primarily addressing the high performance luxury sedan market, turbocharging was believed to be essential and of interest to other developers, an intermediate, naturally aspirated diesel version was not considered.

The 2.4-ℓ BMW diesel is derived from the M 60 gasoline engine which was normally produced in 2.0-ℓ and 2.3-ℓ versions. The engine uses a Ricardo Comet Mark V Swirl chamber, a Bosch rotary type injection pump, and a Garrett AiResearch turbocharger with an integral wastegate. The 2.4-ℓ displacement was obtained by slightly increasing the stroke of the 2.3-ℓ gasoline version. The diesel, weighs 400 lb including the turbocharger, only 22 lb more than the 2.3-ℓ gasoline version. It develops 115 hp at 5000 rpm, and a maximum torque of 137 ft-lb at 2500 rpm. As shown in Figure 3.6-1, the maximum power is slightly less than that of the 2-ℓ gasoline version, but there is considerably more torque available throughout the speed range. In the 323 series cars, the diesel is expected to perform as well as the smaller gasoline version at an average (mpg) fuel savings on the order of 20%.

In a very recent press release (Ref. 31) BMW revealed design features and test data pertaining to a 3.2-ℓ six-cylinder diesel engine with plans for marketing it in the near future with 7-series BMW cars that each weigh about 3700 lb. The development of this engine is a joint effort between BMW, Austrian Steyr Daimler Puch and the Franz List Institute (DVL) in Graz, Austria. It started approximately in 1978, as was implied in earlier press information (Ref. 30).

The design approach taken is unique when compared to all other high speed diesel engines on the market today and scheduled for marketing in the near future. It uses open-chamber, direct fuel injection by means of individual (apparently cam operated) injectors without a central injection pump, and is of a monoblock design, i.e., cylinder block and heads are cast in one

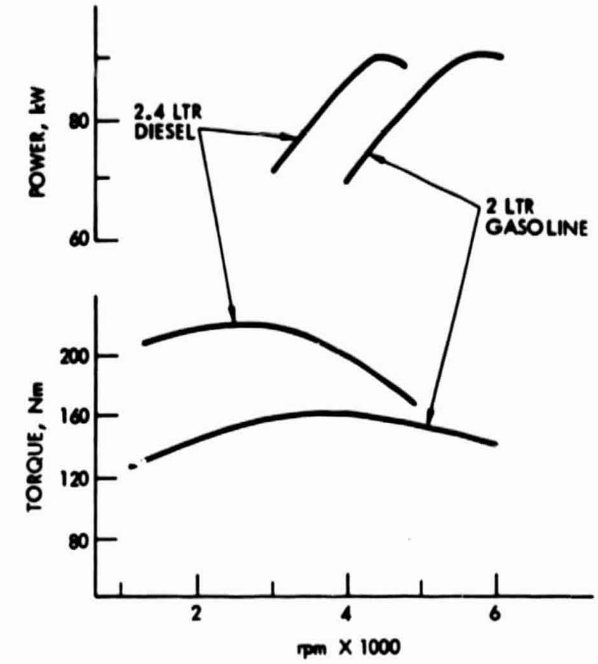


Figure 3.6-1. Comparison of Power and Torque Characteristics for BMW 2-ℓ Gasoline vs. 2.4-ℓ Diesel Engine

piece. The entire engine is fully encapsulated with insulating materials to attenuate unavoidable noise radiation and to reduce thermal losses. The engine is also turbocharged and develops a maximum power of 150 hp.

In tests with a 7-series BMW car, the engine was shown to be capable of accelerating the relatively heavy (4000 lb IW) car from 0 to 60 mph in less than 13 sec, and of yielding 34.5 mpg at a constant speed of 60 mph. Fuel economy of 18.2 mpg was reportedly obtained at 113 mph.

It is interesting to note how the open chamber noise problem was resolved in an apparently satisfactory manner in three separate steps: (1) by eliminating high pressure fuel lines for better control of the injection rate, (2) by stiffening the block and thus reducing the radiation of combustion noises, and (3) by encapsulating the entire power plant to insulate and to confine still unavoidably generated diesel noises within a controlled space. Although none of these measures are innovative by themselves, in concert they represent a uniquely new design approach that may well revolutionize the design of small high-speed diesels. If striking fuel economy advantages can be demonstrated over a longer period of time (at least two model years) that will justify the introduction of costly design features such as the individual injector pumps, and the monoblock design which probably requires high technology foundry techniques, special tooling, and more time for production. Apparently all of the BMW diesel projects are nearing the production stage, starting in 1983 with the above described turbocharged 2.4-ℓ units. A new engine assembly plant jointly owned by BMW and Steyr is being built in Austria that will be ready to produce BMW gasoline engines in 1982,

the 2.4-ℓ diesel engine in 1983, and the newly developed BMW-Steyr 3.2-ℓ open chamber diesel starting in 1984.

Ford Motor Company, reportedly, will be the first U.S. automaker to use BMW-Steyr diesel technology in their line of cars, and is planning to buy 100,00 units per year starting in 1983. The smaller 2.4-ℓ BMW diesel is being considered for use in the Ford Lynx models, and the larger 3.2-ℓ BMW-Steyr diesel is under consideration for Thunderbirds and Cougars. If by 1984 the predicted level of 250,000 to 300,000 engines per year can be established, BMW-Steyr will be a major force among the largest producers of small automotive diesel engines in the free world.

3.7 COMPARISON OF PROMINENT FEATURES

A comparison of performance criteria from the consumer standpoint is difficult because, as can be seen from Table 3.7-1, each of the discussed products has been developed with entirely different marketing objectives in mind. Based upon sales figures, all of the listed vehicles have met their marketing objectives extremely well, although technical differences do exist which will be a decisive factor for survival against competition and future EPA emission constraints.

Table 3.7-2 shows vehicular fuel economy and performance criteria side-by-side as discussed in the foregoing paragraphs. Plotting composite fuel efficiencies against vehicle weight (Figure 3.7-1), it can be seen that vehicle weight is the primary factor that determines fuel efficiencies. Plotting transport efficiencies in terms of transport work per fuel consumed (lb x mpg) against vehicle weight (Figure 3.7-2) or fuel efficiency (Figure 3.7-3) it becomes apparent that transport efficiency is primarily the composite effect of vehicle weight, engine efficiency, drive train selection and parasitic losses, such as aerodynamic drag, rolling resistance, etc. All of these factors have been highly optimized according to the state of-the-art in all of the discussed cases. Transport efficiency therefore, varies only very little with vehicle weight or fuel efficiencies achieved. As indicated by the triangular symbols, turbocharging obviously has improved vehicle fuel efficiency and on transport efficiency (mi-lb/gal) by approximately 10% in all cases.

Comparing the installed power-to-weight ratio to the production year, Figure 3.7-4 shows a trend toward higher power-to-weight ratios during recent years to improve diesel performance. According to Figure 3.7-5, which shows the relationships between power-to-weight ratio and acceleration time, a near Otto-equivalent performance level has actually been achieved with turbocharging in the case of Volkswagen (VW) and Mercedes Benz (MB). The BMW 732 D will be the first high performance diesel car on the road exceeding most gasoline-powered cars in driveability. Turbocharging generally results in a lower acceleration time than does increasing only the displacement of a naturally aspirated engine, provided the power-to-weight ratio remains unchanged. This is attributed to the use of a waste gate, which curbs power to protect the engine without losing the advantage of increased low end torque.

Table 3.7-1. Comparison of Design and Marketing Objectives

Producer	Car Model	Weight lbs	Passenger Capacity	Design Objective	Market Addressed
Volkswagen	Rabbit (Golf)	2250	4	Max. economy; no frills transportation	Primarily economy minded buyer
Peugeot	504D	3500	4 (5)	Economy, comfortable transportation	Economy minded but more affluent buyer
Daimler Benz	300D	4000	5	Economy, comfortable transportation	Economy but prestige minded buyer
	300SD	4000	5	Economy, high-speed touring and comfort	Power and economy prestige minded
General Motors	Olds	4500	6	Economy with standard car comfort and capacity	Economy but space and comfort minded buyer
	Cadillac Seville	4500	5 (6)	Economy with standard car luxury and capacity	Economy, luxury and prestige minded buyer

Table 3.7-2. Comparison of Vehicle Performance Criteria

Producer		Daimler Benz		Peugeot			Volkswagen		General Motors Olds Division	BMW	
Model		300 D	300 SD	504 D	504 D	506 D	Rabbit	Rabbit	Delta 88 D	524 D	732 D
Production Year		1977	1978	1974	1977	R&D	1977	R&D	1978	1982	1984
Nominal Power, hp		11	110	62	71	73	50	71	120	115	150
Inertia Weight, lbs		4000	4000	3500	3500	3500	2250		4500	2830	3975
Power/Weight Ratio x 10^3		19.3	27.5	17.7	20.3	20.9	22.2	2250	26.7	31.8	37.7
Comb. Chamber Concept		Pre	Pre	Swirl	Swirl	Swirl	Swirl	Swirl	Swirl	Swirl	Dir *
Charge System		NA	TC	NA	NA	TC	NA	TC	NA	TC	TC *
Transmission		Auto.	Auto.	4-Spd	4-Spd	5-Spd	4-Spd	5-Spd	Auto.	N/A	N/A
Drive		Rear	Rear	Rear	Rear	Rear	Front	Front	Rear	Rear	Rear
0-60 Acceleration, sec.		21.0	13.5	23.6	21.0	17.5	18.5	13.5	16.0	N/A	13
Fuel Economy -	City	22.7	23.7	27.0	27.0	27.0	39.0	51.0	21.0	N/A	N/A
(mpg)	Hwy.	27.5	28.8	35.0	32.0	32.0	52.0	63.0	30.0	N/A	24.5
	Comb.	24.5	25.8	30.0	29.0	29.0	44.0	56.0	24.0	N/A	N/A
Emission	HC	.29	.17	.91	.7		.16	.11	.64	N/A	N/A
(g/mi)	CO	1.0	.8	2.0	1.8		1.0	.8	1.5	N/A	N/A
	NO_x	1.7	1.85	.96	1.02		1.2	.9	1.62	N/A	N/A
	Partic.	.49	.5		.38		.29		.92	N/A	N/A
Interior Noise Level (db)		74	74	78	78		72		70.5	N/A	N/A

*Abbreviations: Pre - Prechamber; Swirl - Ricardo Swirlchamber; Dir - Direct injection open chamber; NA - Naturally Aspirated; TC - Turbocharged; N/A - Not applicable

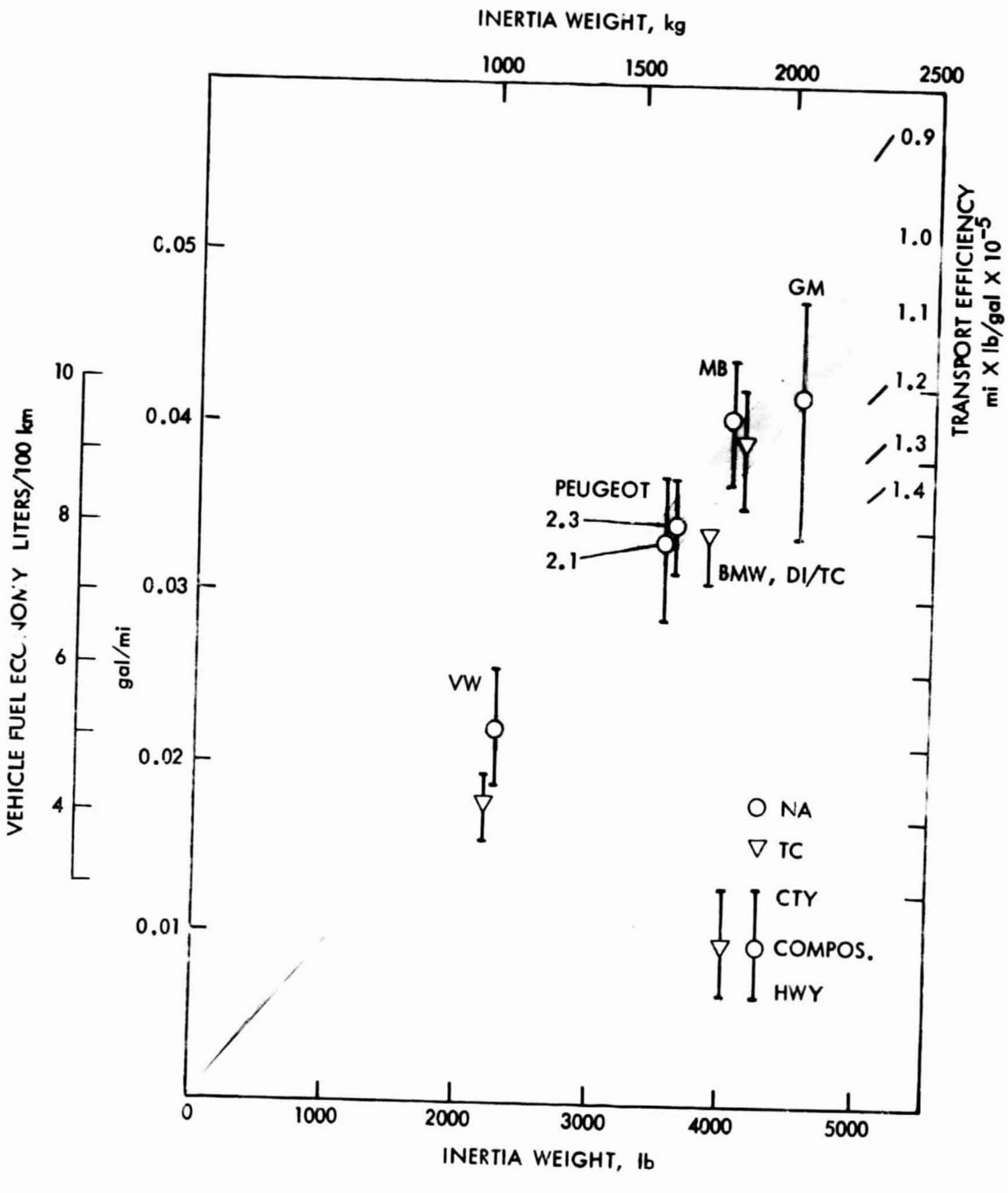


Figure 3.7-1. Fuel Economy vs. Vehicle Inertia Weight

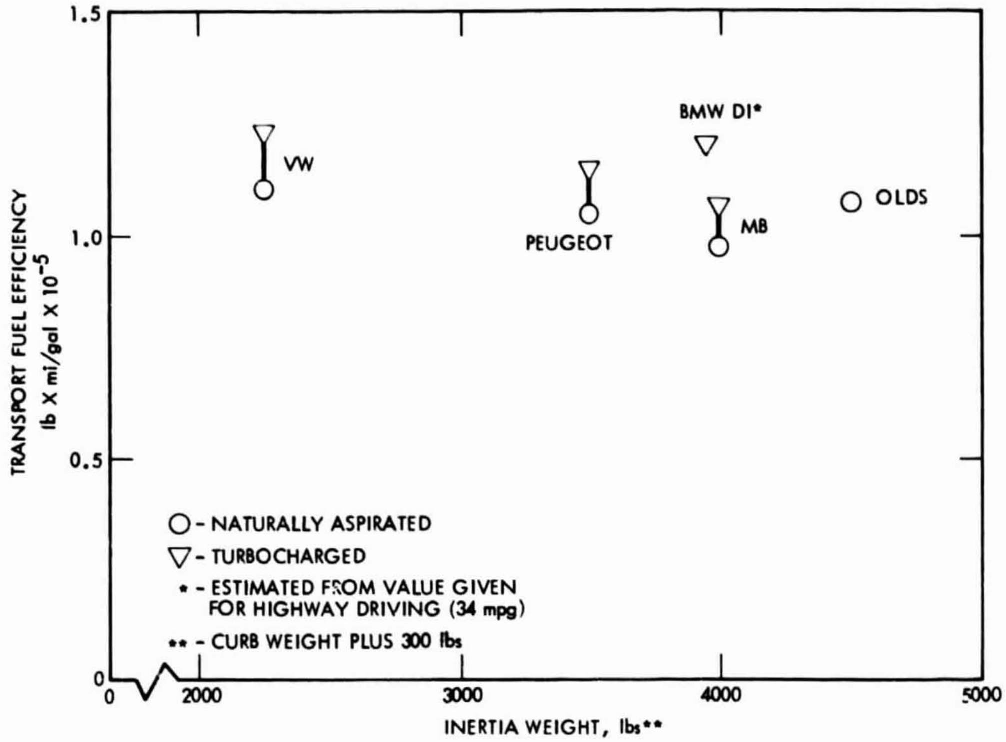


Figure 3.7-2. Effect of Vehicle Inertia Weight on Transport Efficiency

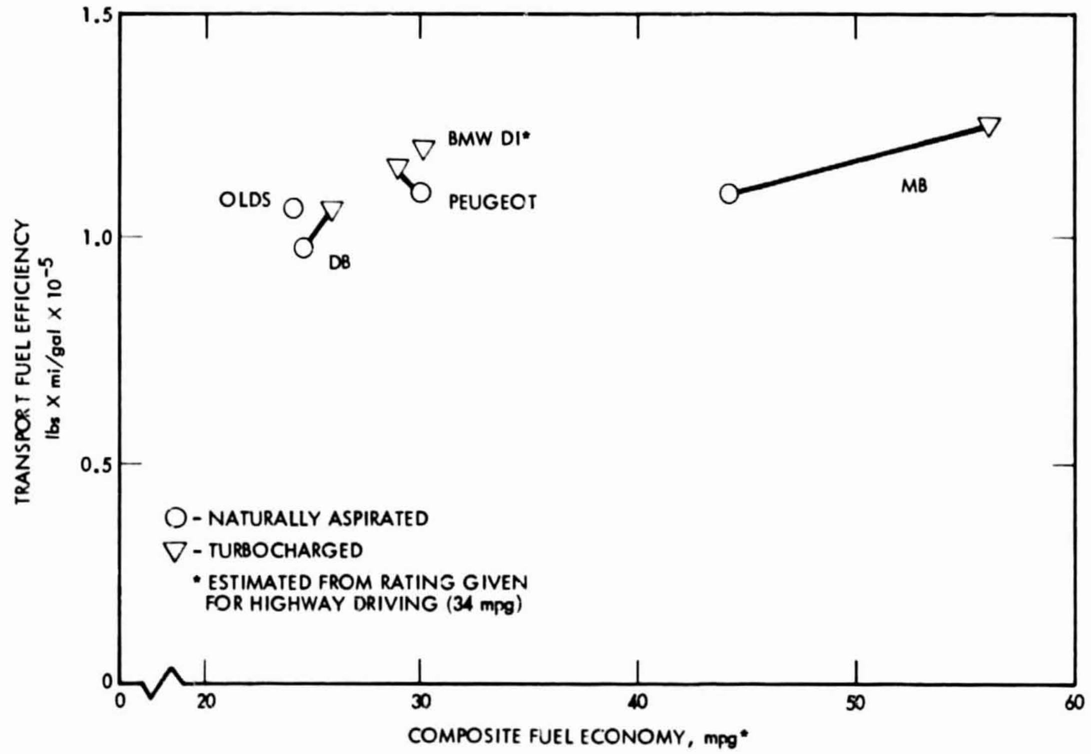


Figure 3.7-3. Vehicle Fuel Economy vs. Transport Efficiency

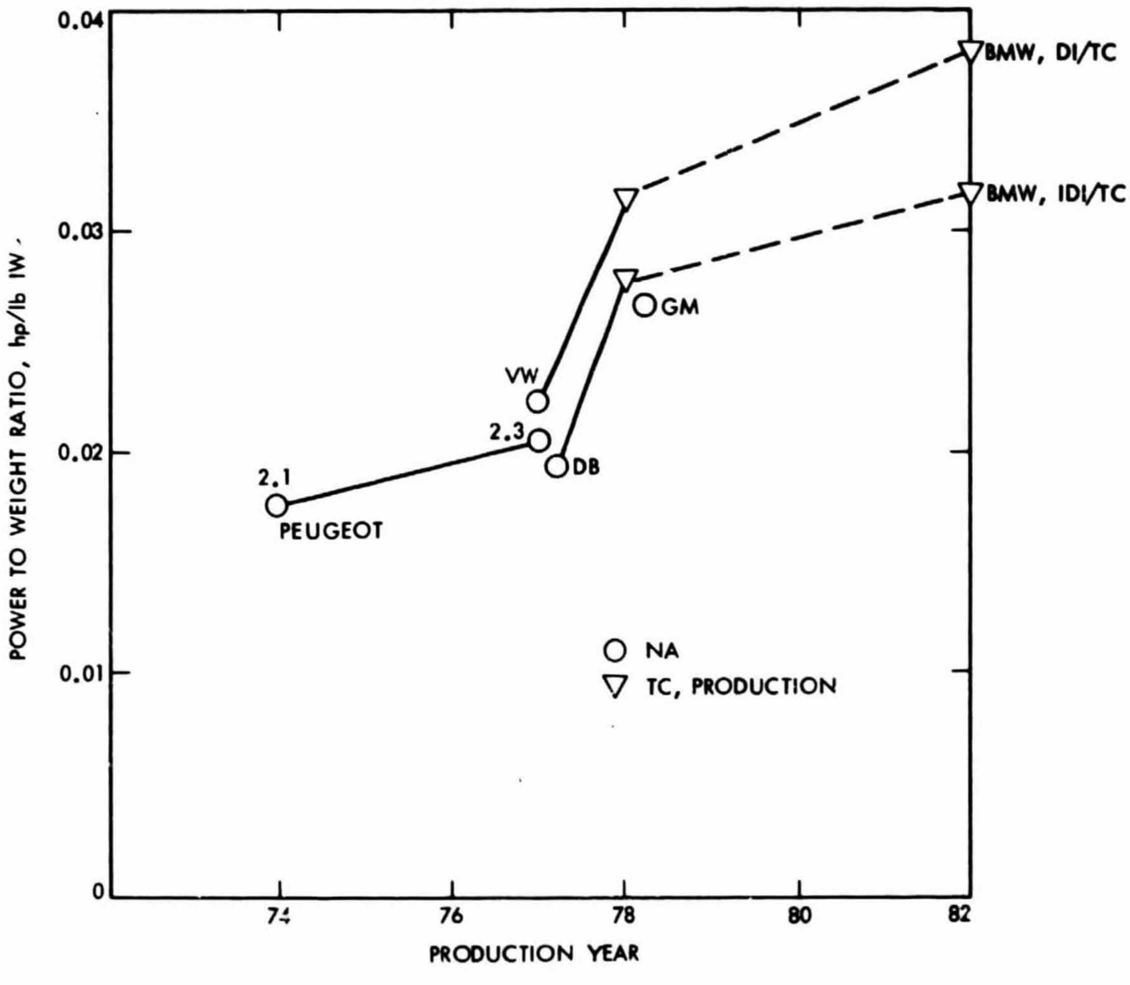


Figure 3.7-4. Power-to-Weight Ratio vs. Production Year

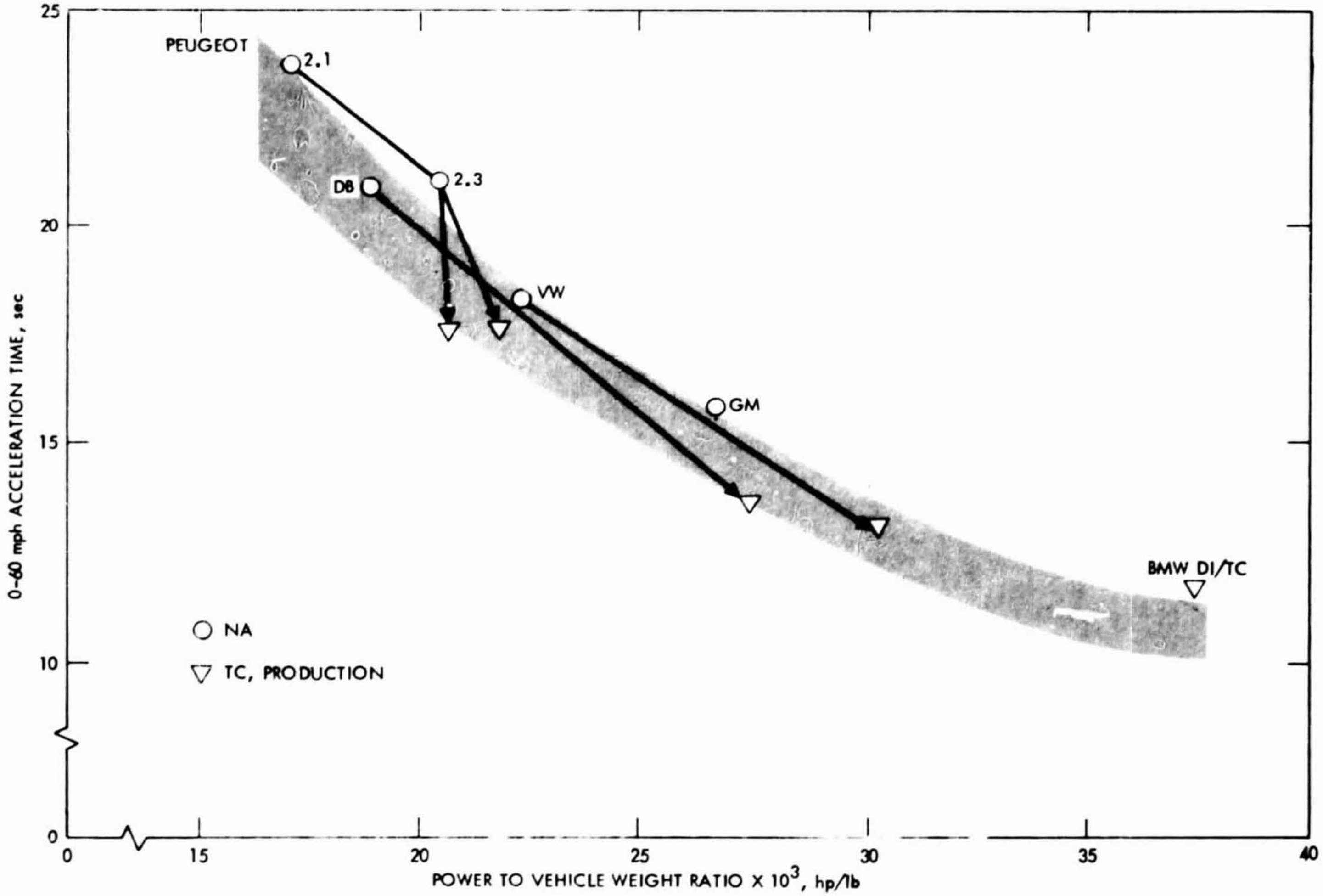


Figure 3.7-5. Power-to-Weight Ratio vs. 0-60 mph Acceleration Time

Figure 3.7-6 shows the relationships between vehicle weight and NO_x emission per mile driven. Vehicle weight is definitely a primary factor, but differences in system quality and installed power also have a strong and compounding effect. Peugeot (first) and Olds (second) have the lowest NO_x emission in terms of g/mi, as well as g/mi x lb. A low power-to-weight ratio (Figure 3.7-5) and a clean engine are the primary reasons for lowest emission levels. Volkswagen leads the field with regard to absolute g/mi NO_x which is brought about by a low vehicle weight. Otherwise VW ranks behind Peugeot, Oldsmobile and Daimler Benz, when grouped by quality in terms of g/mi x lb. Turbocharging can have a NO_x reducing effect (see VW in Figure 3.7-6) depending upon engine characteristics, turbocharger engine match, waste gate control schedules and gearing.

As can be seen from Figure 3.7-7, vehicle weight also has a strong effect on particulate emission as with NO_x. Volkswagen has the lowest particulate emission in absolute terms because of low vehicle weight. Peugeot and Mercedes Benz are best in quality in terms of g/mi and vehicle weight. GM/Olds is highest in terms of g/mi because of high vehicle weight and relatively poor emission quality in terms of grams per transport work (g/lb x mi). Reportedly the Olds diesel has been cleaned up to meet the 0.6 g/mi EPA particulate standard.

It is expected that further developments in the near future will iron out existing differences, thus producing a smaller family of engine designs that exhibit optimum properties in regard to clean combustion, weight and performance. Vehicle weight and acceleration capability will then be the primary factors that determine emission characteristics. To meet future projected EPA NO_x and particulate emission standards without the waiver, and with the lowest acceptable power, a reduction of vehicle weights below that of the present VW Rabbit will be necessary. With the improvement potential envisioned for the near future, VW and Peugeot seem capable of staying in the market without major improvements, provided the waiver from the 0.2 g/mi standards for particulates will be extended. Daimler Benz and Oldsmobile will stay in the market for the duration of the 1.5 g/mi NO_x waiver, but as is evident from Figure 3.7-6 both will have problems in meeting NO_x and particulate standards after 1982. With regard to interior noise (Table 3.7-2), Peugeot is obviously the noisiest, and General Motors/Oldsmobile is best. Differences in injection pressure and body insulation are responsible for the observed 7.5 db difference.

Table 3.7-3 A through E summarizes typical engine design and performance criteria as available from published material. One of the major differences between engines is the use of various combustion chamber concepts. Daimler Benz uses their trusted and proprietary pre-chamber, and all others currently marketed are committed to the use of a Ricardo type swirl chamber of their own design. Only BMW-Steyr has taken a daring step towards the open chamber in their on-going development.

All of these diesels are sharply divided into two groups, each characterized by an entirely different design background and approach - the diesel and the gasoline engine diesel conversion. Daimler Benz and Peugeot are diesels designed from scratch to be from the outset a diesel, whereas Volkswagen and General Motors/Oldsmobile are diesel conversions based on existing

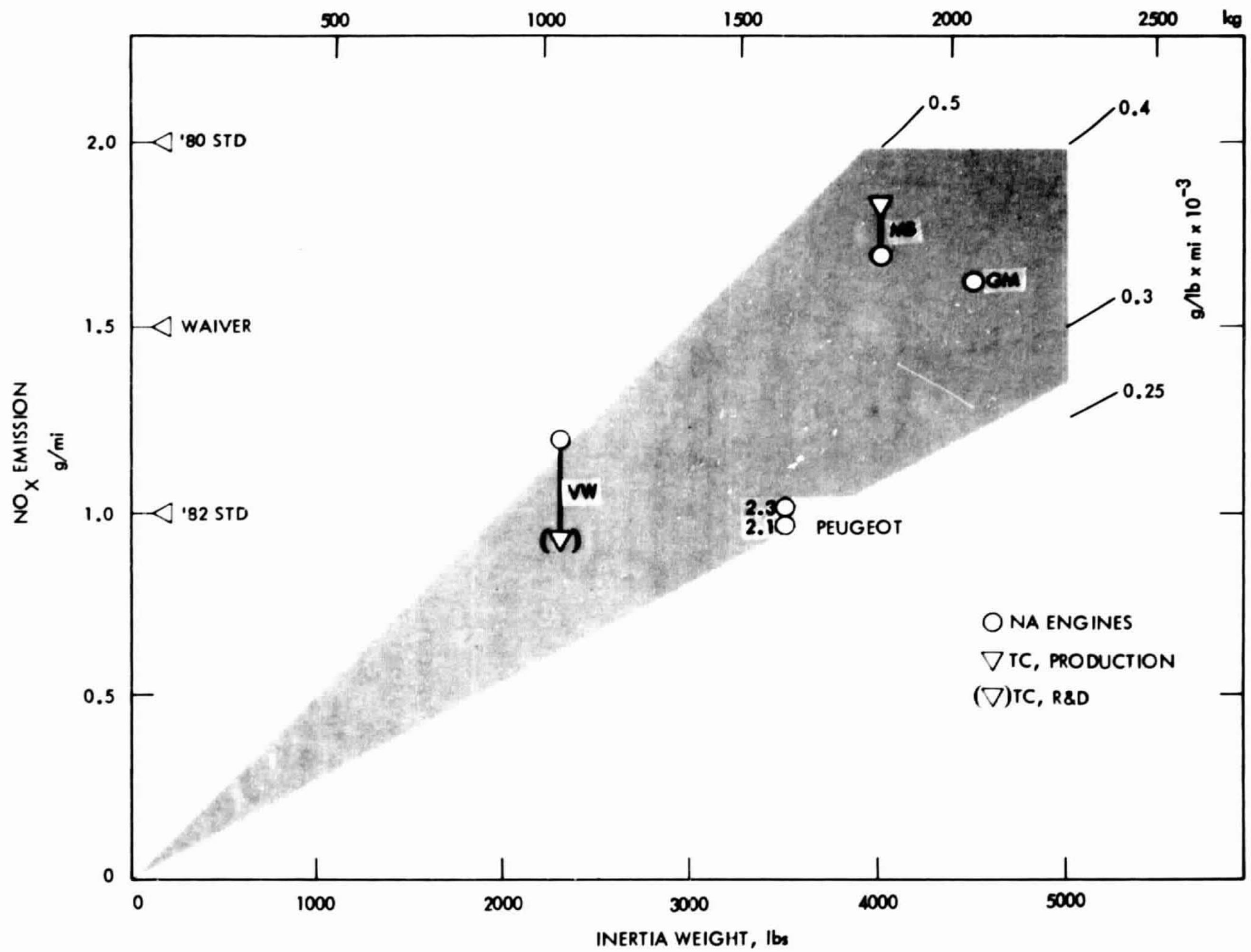


Figure 3.7-6. Vehicle Weight vs. No_x Emission

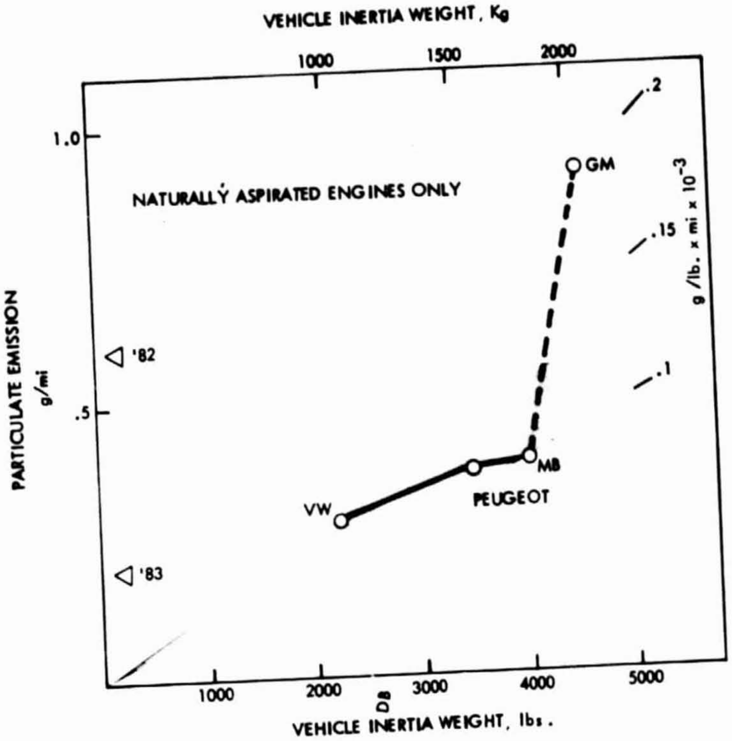


Figure 3.7-7. Vehicle Weight vs. Particulate Emission

gasoline engines. With typical design features such as six head bolts around cylinders instead of the usual four, machined cooling passages between cylinders, oil cooled pistons, and sodium cooled outlet valves, the Daimler Benz OM 612 "A" engine is the most conservatively designed, but also the most expensive turbocharged automotive diesel on the market.

Peugeot has broken with conservative diesel design practice by using an aluminum cylinder head and a larger bore to stroke ratio on their 1977 models. This was apparently necessary to compensate for a slightly over-designed 504 block which, in contrast to all others, has inserted wet cylinder sleeves. This has given Peugeot a slight edge over Daimler Benz with regard to engine power-to-weight ratio, apparently without other noticeable sacrifices. Peugeot and Daimler Benz automotive diesels have established a service and reliability record that is comparable to that of truck diesels. Volkswagen has obviously taken the initiative with a daring but well optimized diesel conversion approach which has put them ahead of the field in power per unit displacement, structural weight and price.

Table 3.7-3A. Summary of Engine Criteria, Daimler Benz

Characteristic	Daimler Benz	
Engine, Production Year	300D, 1977	300D, 1978
Derivation	Conv. Diesel Design	
Displacement, CID (liter)	184 (3)	184 (3)
No. of Cylinders & Arrangement	5 in line	5 in line
Installation	Longitudinal	
Charge Sys./Pres. Boost, Bar	NA	TC/1.7
Bore/Stroke, in. (mm)	3.58/3.64	(91/92.4)
Block Material/Cylinder	Cast Iron Integral[a]	
Head Material	Cast Iron	
Valve Mechanics	OHC[b]	
Cam Drive	Chain	
Piston Material	Aluminum Alloy[c]	
No. of Crankshaft Bearings	6	6
Compression Ratio	22	21.5
Chamber Type	Pre-Chamber	
Injection System-Pump Type	Individual In Line Plungers	
Pressure, psig	1800	2099
Engine Weight, lb (kg)	515 (234)	548 (249)
Nom. Power, hp/Eng. Speed, rpm	77/4200	110/4200
Max. Torque, ft-lb/Eng. Speed, rpm	115/2400	168/2400
Eng. Wt., lb/Nom. Power/hp	6.68	4.98
Bore Stroke Ratio	0.98	0.98
Nom. Power, hp/Eng. Wt., lb (kg)	0.149 (.328)	0.190 (.418)
Nom. Power, hp/Eng. Displ., ℓ	25.7	36.7
Max. Torque, ft-lb/Eng. Displ., ℓ	38.3	56.0
Max. Torque-to-Power Speed Ratio	0.57	0.57
Piston Speed at Max Power, ft/min	2548	2548
Piston Speed at Max Power, m/s	12.9	12.9

Legend: n/a = not available NA = naturally aspirated
TC = turbocharged OHC = overhead cams

Remarks: [a]coding slots open to head gasket between cylinders
[b]sodium cooled exhaust valves
[c]oil cooled pistons

Table 3.7-3B. Summary of Engine Criteria, Peugeot

Characteristic	Peugeot		
Engine, Production Year	504D, 1974	504D, 1976	506D R&D
Derivation	Conv. Diesel Design		
Displacement, CID (liter)	129 (2.1)	141 (2.3)	141 (2.3)
No. of Cylinders & Arrangement	4 in. line	4 in. line	4 in. line
Installation		Longitudinal	
Charge Sys./Pres. Boost, Bar	NA	TC/1.7	TC/1.6
Bore/Stroke, in. (mm)	3.6/3.2 (91.5/83)	3.7/3.2 (92.7/83)	n/a
Block Material/Cylinder	Cast Iron/Wet Sleeves		
Head Material	Aluminum Alloy		
Valve Mechanics	OH - Push Rods		
Cam Drive	Chain	Chain	Chain
Piston Material	Aluminum Alloy		
No. of Crankshaft Bearings	5	5	5
Compression Ratio	22.2	22.4	21
Chamber Type	Ricardo Swirl		
Injection System-Pump Type	Individual In Line Plungers		
Pressure, psig	n/a	n/a	n/a
Engine Weight, lb (kg)	415 (188)	415 (188)	441 (200)
Nom. Power, hp/Eng. Speed, rpm	62/4500	71/4500	73/4150
Max. Torque, ft-lb/Eng. Speed, rpm	91/2400	99/2400	137/2000
Eng. Wt., lb/Nom. Power/hp	6.69	5.8	6.04
Bore Stroke Ratio	1.08	1.12	n/a
Nom. Power, hp/Eng. Wt., lb (kg)	0.149 (0.328)	0.171 (0.377)	0.166 (0.366)
Nom. Power, hp/Eng. Displ., ℓ	29.5	30.8	31.7
Max. Torque, ft-lb/Eng. Displ., ℓ	42.8	43.0	59.6
Max. Torque-to-Power Speed Ratio	0.53	0.53	0.48
Piston Speed at Max Power, ft/min	2255	2451	2451
Piston Speed at Max Power, m/s	11.5	12.5	12.5

Legend: n/a = not available NA = naturally aspirated
TC = turbocharged OHC = overhead cams

Table 3.7-3C. Comparison of Engine Criteria, Volkswagen

Characteristic	Volkswagen	
Engine, Production Year	Rabbit, 1978	Rabbit, R&D
Derivation	Gasoline Conversions	
Displacement, CID (liter)	90 (1.5)	90 (1.5)
No. of Cylinders & Arrangement	4 in. line	4 in. line
Installation	Transverse	
Charge Sys./Pres. Boost, Bar	NA	TC - n/a
Bore/Stroke, in. (mm)	3.01/3.15	(76.5/80)
Block Material/Cylinder	Cast Iron/integral	
Head Material	Aluminum	
Valve Mechanics	OHC	
Cam Drive	Cogged Belt	
Piston Material	Aluminum	Aluminum
No. of Crankshaft Bearings	5	5
Compression Ratio	23	n/a
Chamber Type	Ricardo Swirl	
Injection System-Pump Type	Rot. Distributor	
Pressure, psig	n/a	n/a
Engine Weight, lb (kg)	286 (125)	292 (133)
Nom. Power, hp/Eng. Speed, rpm	50/5000	70/5000
Max. Torque, ft-lb/Eng. Speed, rpm	61/3000	91/3000
Eng. Wt., lb/Nom. Power/hp	5.72	4.17
Bore Stroke Ratio	0.96	0.96
Nom. Power, hp/Eng. Wt., lb (kg)	0.175 (0.385)	0.239 (0.526)
Nom. Power, hp/Eng. Displ., ℓ	33.3	46.6
Max. Torque, ft-lb/Eng. Displ., ℓ	38.1	66.6
Max. Torque-to-Power Speed Ratio	0.60	0.60
Piston Speed at Max Power, ft/min	2625	2520
Piston Speed at Max Power, m/s	13.3	12.8

Legend: n/a = not available; TC = turbocharged; NA = naturally aspirated; OHC = overhead cams

Table 3.7-3D. Comparison of Engine Criteria, General Motors - Oldsmobile

Characteristic	GM
Engine, Production Year	Olds, 1978
Derivation	Gasoline Conversions
Displacement, CID (liter)	350 (5.7)
No. of Cylinders & Arrangement	V8
Installation	Longitudinal
Charge Sys./Pres. Boost, Bar	NA
Bore/Stroke, in. (mm)	4.057/3.385
Block Material/Cylinder	Cast Iron/Integral
Head Material	Cast Iron
Valve Mechanics	OH Pushrods
Cam Drive	Chain
Piston Material	Aluminum Alloy
No. of Crankshaft Bearings	5
Compression Ratio	22.5
Chamber Type	Ricardo Swirl*
Injection System-Pump Type	Rot. Distributor
Pressure, psig	n/a
Engine Weight, lb (kg)	n/a
Nom. Power, hp/Eng. Speed, rpm	120/3600
Max. Torque, ft-lb/Eng. Speed, rpm	220/1600
Eng. Wt., lb/Nom. Power/hp	n/a
Bore Stroke Ratio	1.2
Nom. Power, hp/Eng. Wt., lb (kg)	n/a
Nom. Power, hp/Eng. Displ., ℓ	21.0
Max. Torque, ft-lb/Eng. Displ., ℓ	38.5
Max. Torque-to-Power Speed Ratio	0.44
Piston Speed at Max Power, ft/min	2030
Piston Speed at Max Power, m/s	10.3

Legend: n/a = not available NA = naturally aspirated
TC = turbocharged OHC = overhead cams
Remarks: *Swirl, own design, angled throat, flame slot.

Table 3.7-3E. Comparison of Engine Criteria, BMW

Characteristic	BMW
Engine, Production Year	524D, 1982
Derivation	Gasoline Conversions
Displacement, CID (liter)	2.4
No. of Cylinders & Arrangement	4 in line
Installation	
Charge Sys./Pres. Boost, Bar	
Bore/Stroke, in. (mm)	
Block Material/Cylinder	n/a
Head Material	n/a
Valve Mechanics	n/a
Cam Drive	n/a
Piston Material	n/a
No. of Crankshaft Bearings	5
Compression Ratio	
Chamber Type	Ricardo Swirl
Injection System-Pump Type	n/a
Pressure, psig	
Engine Weight, lb (kg)	400 (182)
Nom. Power, hp/Eng. Speed, rpm	115/5000
Max. Torque, ft-lb/Eng. Speed, rpm	137/2500
Eng. Wt., lb/Nom. Power/hp	3.48
Bore Stroke Ratio	n/a
Nom. Power, hp/Eng. Wt., lb (kg)	0.286
Nom. Power, hp/Eng. Displ., ℓ	47.9
Max. Torque, ft-lb/Eng. Displ., ℓ	57
Max. Torque-to-Power Speed Ratio	0.5
Piston Speed at Max Power, ft/min	n/a
Piston Speed at Max Power, m/s	n/a

Legend: n/a = not available NA = naturally aspirated
TC = turbocharged OHC = overhead cams

The outstanding fuel efficiency of the Rabbit engine can be credited to an optimum cylinder size (Figure 3.7-8), and to an efficient overhead cam and belt drive, in conjunction with a tuned inlet and exhaust system that permits high engine speed with relatively low volumetric losses. The incorporation of the latest state-of-the-art in injection and combustion systems have also contributed to the outstanding high speed capability of the VW engine.

Compared to VW and BMW, General Motors/Oldsmobile has taken a relatively conservative approach where cylinder loading (Figure 3.7-8) and piston speed are concerned. Combustion and injection pressures are relatively low, but this may well be responsible for the relatively high particulate emission level (Figure 3.7-7) typical for this engine. Volume puts the engine at the low end of the field with respect to power to displacement and power to engine weight ratio which, as can be seen from Figure 3.7-9, are practically in linear relationship to each other for state-of-the-art designs.

As with VW, the BMW 2.4-ℓ diesel conversion also has an optimum cylinder displacement layout (Figure 3.7-8) and leads the entire field in specific engine weight (Figure 3.7-9). It will be the first high power and practically Otto-engine equivalent diesel on the market.

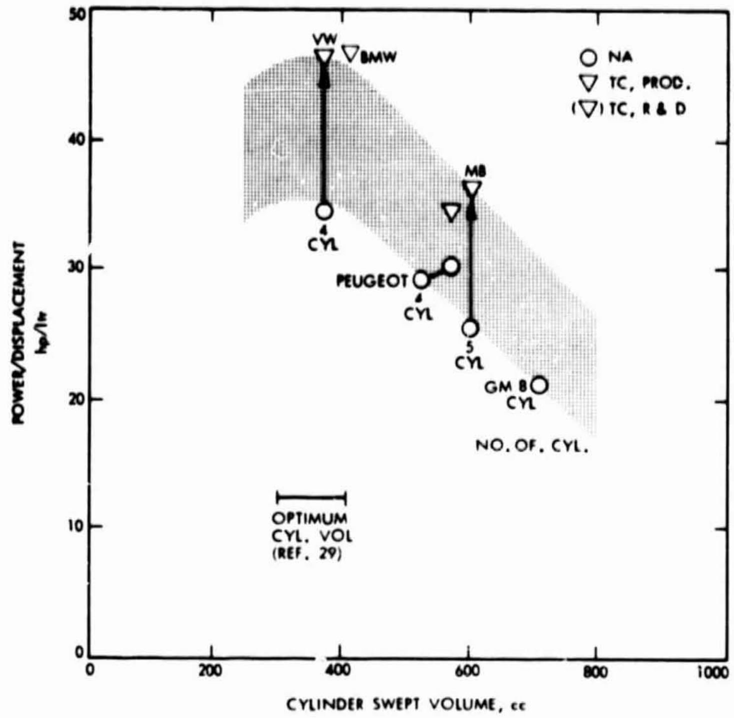


Figure 3.7-8. Cylinder Volume vs Power Output for Unit Displacement

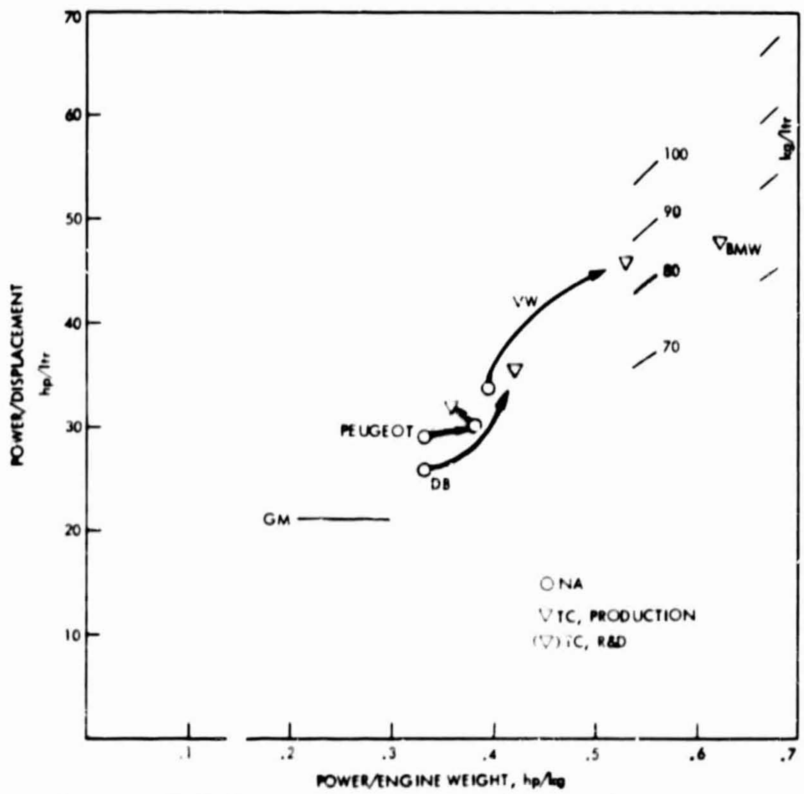


Figure 3.7-9. Comparison of Engine Performance Size and Weight Relationships

SECTION 4

APPROACHES TO THE EMISSION PROBLEM

4.1 BACKGROUND AND SCOPE

The automotive diesel still has a great potential for further improvement in all areas, including that of emissions. The diesel-oriented community is making large investments in R&D work to further improve engine performance, and to resolve the emission problem to the best of its ability. A variety of government and privately sponsored R&D projects are in progress with the major emphasis on engine emission improvements, and on the study of potentially adverse effects on health resulting from diesel exhaust. The following paragraphs present data and approaches that appear to be most favorable for diesel emission improvement.

Diesel engines emit a variety of pollutants, including hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NO_x), oxides of sulfur, aldehydes, particulates, polymer aromatics (PNA's), and odor. The relationship between these pollutants and certain physical characteristics and phenomena are fairly well known. However, the physical/chemical interaction leading to specific species of pollutants is still widely unexplored, especially where the formation of particulates is concerned. Therefore, the realistic modeling and the quantitative determination of pollutants becomes a difficult task.

A wealth of related literature has been published in recent years. Drawing primarily from References 8 and 32, which can be considered the best condensed collections of emission-related data, attention is mainly centered around the problem emissions, NO_x, and particulates. Only those physical facts and interactions will be discussed that are of primary interest to the engine designer and are necessary for the understanding of remedial measures and new design approaches.

Except for NO_x, all of the emissions that are currently regulated or will be regulated in the near future (Table 2.3-1), are in some way the result of imperfect and incomplete combustion. The primary causes are insufficient fuel penetration and atomization, inadequate turbulence and mixing, the quenching of oxidation reactions near cold walls, and/or after-injection at a point where combustion cannot be completed.

The formation of NO_x is kinetically controlled and increases with temperature, oxygen concentration and residence time of the hot gases under high temperature conditions. Therefore, the NO_x emissions from pre-chamber engines are lower than those of existing direct injection or open chamber engines. In pre-chamber engines, primary combustion takes place in a confined space under fuel-rich conditions without excess oxygen, and very little NO_x is formed in the pre-chamber. Excess oxygen then becomes available in the main chamber to complete combustion, but temperatures are lower and NO_x formation is rapidly minimized.

C-2

The emission of particulates and smoke is also closely related to combustion efficiency. The formation of the organic compounds, which are of primary concern, is not fully understood. Reactions among the various compounds occur within the combustion chamber and continue throughout the exhaust system, including the measuring system, such as the dilution tunnel and filters. This makes conclusive measurement and characterization very difficult. Depending mainly on engine quality and vehicle weight (Figure 4.1-1) the particulate emission from diesel automobiles of the sizes studied ranges typically from 0.3 to 1. 'mi, compared to 0.3 g/mi for uncontrolled gasoline powered cars, and abou. '1 g/mi for cars equipped with catalytic converters, and using unleaded gasoline. The EPA schedule (Table 2.3-1) requires a reduction to 0.6 g/mi in 1981 and 1982, and to 0.2 g/mi from 1982 on. While the compliance with a 0.6 g/mi standard is possible with present technology, the achievement of particulate emission rates below that is still an unresolved problem.

Systematic studies and characterization tests conducted in various places during recent years have been very successful in identifying the relationship between certain design features, operational factors and pollutants. However, because of the large number of variables involved, the conclusions drawn with one design are not always applicable to another. An extensive characterization of each individual design will still be necessary to obtain quantitatively conclusive results. The inherent tradeoff between NO_x, engine efficiency and related pollutants (smoke, particulates, etc.) makes the development of future diesel engines very difficult, time consuming and costly. The engine designer must find the right combination of design features and measures that, in absolute terms, satisfies the projected emission requirements with a reasonable or prescribed margin, while simultaneously keeping fuel efficiency penalties to an absolute minimum. The consensus amoung diesel experts is that most current diesel engine designs can be modified to meet the emission requirements for 1981 in regard to gaseous pollutants. Compliance with the EPA scheduled standards for gaseous pollutants will require the introduction of EGR, with attendant fuel efficiency penalties and reliability problems. The controlling of particulates requires new technology not available at this time. The approaches under consideration concentrate primarily on measures to: (1) avoid operations that are most polluting, (2) lower the combustion temperatures, (3) optimize fuel injection schedules, (4) improve combustion efficiency, (5) reduce excess oxygen, (6) entrap particulates, and (7) develop fuel additives that inhibit the formation of particulates. The effectiveness of these measures are discussed in the following paragraphs.

4.2 OPERATIONAL RESTRICTIONS

Operational restrictions are measures that can be implemented without changing the design of an existing engine, by adjusting or modifying injection timing devices and governors as necessary to avoid operating conditions that produce the most pollution. Most of the smoke, odor, and particulates are generated during low-speed, high-torque operations, because of incomplete combustion brought about by fuel-rich mixture, poor internal turbulence and mixing, reduced injection pressure and impaired droplet distribution.

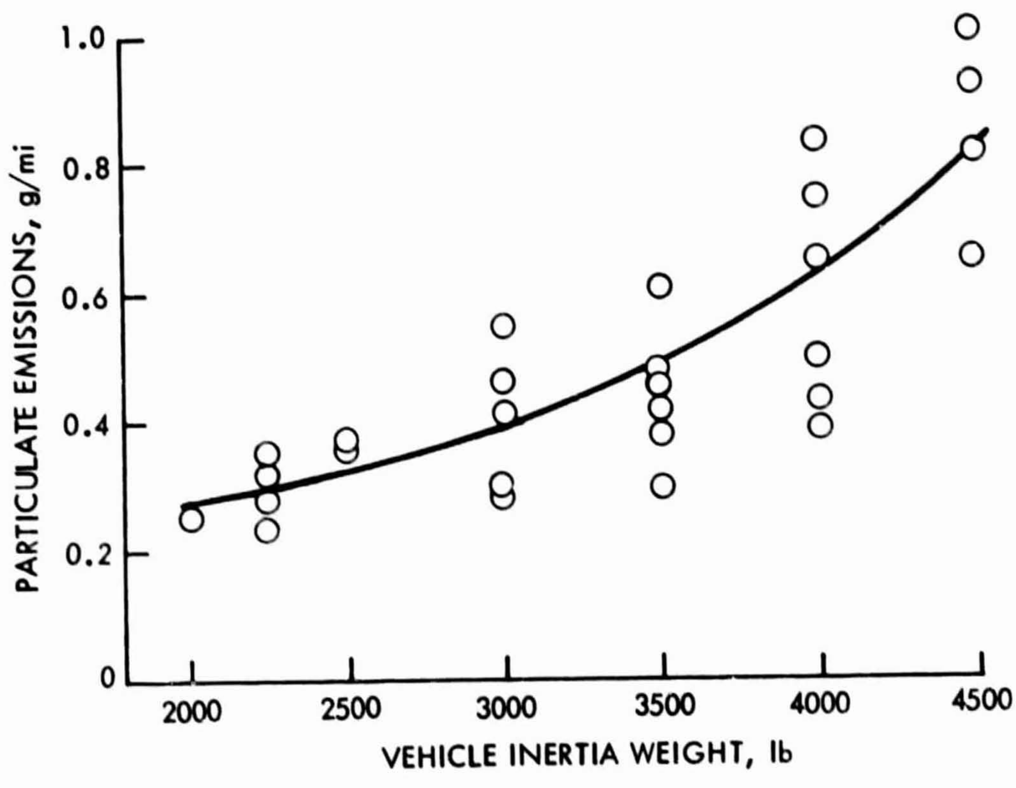


Figure 4.1-1. Urban Cycle Particulate Emissions (Ref. 34)

While operational restriction, commonly referred to as "engine derating", can reduce particulates and NO_x in some engines, related increases in engine specific weight and cost-per-unit power output make approaches in this direction very undesirable.

4.3 INLET AIR TREATMENT

A reduction in NO_x can be obtained by lowering the engine inlet temperature because of the associated reduction in combustion temperature. However, this is meaningful only for turbocharged engines. For example, as shown in Figure 4.3-1, a NO_x reduction on the order of 40% could be obtained if the compressor end temperature is lowered from 265° to 50°F. This was accomplished in the case considered by means of a laboratory water cooler. Data from similar tests have shown that a reduction in engine inlet temperature from 250° to 150°F, which is feasible in an automobile by means of an air-to-air cooler, can still result in a NO_x reduction on the order of 20 to 30%, while simultaneously reducing HC, CO and smoke. Engine fuel consumption was also reduced to a small extent.

Engine inlet air cooling, which is frequently referred to as "after" or "intermediate cooling", has a limited potential for NO_x abatement in automotive engines because of the weight, bulk and cost of the cooling systems. Experience to date tends to indicate that diesel NO_x emissions can be reduced by about 0.2 to 0.3% per degree Fahrenheit of temperature reduction, while simultaneously improving the specific fuel consumption by up to 0.04%, per degree Fahrenheit.

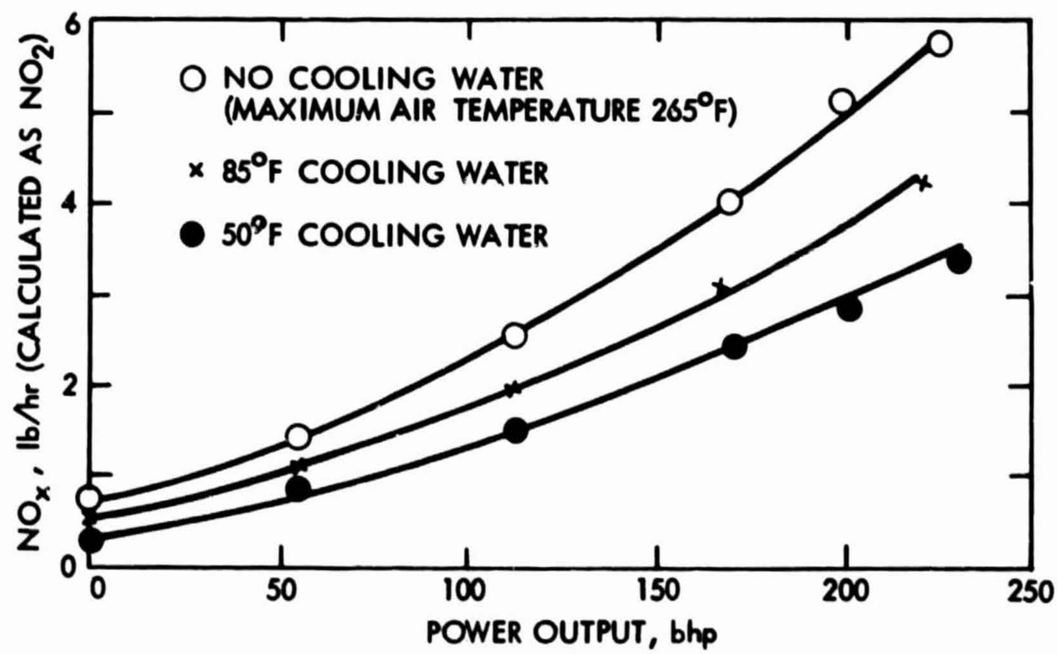


Figure 4.3-1. Effect of Intake Air Temperature on NO_x Emission of a Turbocharged, Open Chamber Diesel Engine at Rated Speed (Ref. 32)

Water added to the intake air results in lowering compression end and combustion temperatures and at the same time can reduce NO_x emissions, as shown in Figure 4.3-2. As long as the amount of water is small, the ignition characteristics of the charge do not change significantly. Large amounts of water result in a considerable ignition delay which requires compensation through advanced fuel injection timing. Without the proper timing correction, the NO_x reducing effects of water are not realized and, as tests have shown, water induction into the inlet air can even lead to an increase of NO_x emission.

Volkswagen has experimented with water injection in their turbocharged Rabbit engine, with a water-to-fuel ratio of up to 40%, with an observed increase of full load mean effective pressure of 6%, and a 5% improvement in specific fuel consumption (Ref. 23). A noticeable reduction in NO_x was also observed but was not enough to ensure compliance with the goal of 0.7 g/mi for prototypes. Although the results obtained were encouraging, VW feels that the need for a dual fuel system and associated problems of icing and corrosion make this approach very unattractive.

Another method known as "fumigation" involves the injection of fuel into the intake air, which tends to reduce the initial heat release rate and associated chamber pressure rise and noise generation. Fumigation has also been found to reduce NO_x but showed an increase in CO, HC and smoke emission.

4.4 INJECTION SYSTEM

Refinements of injection timing and nozzle design are first-choice approaches toward improving the emission characteristics of diesel engines, because they are feasible without changing the existing basic design. Injection timing has a strong effect on combustion peak temperatures and attendant NO_x formation, particularly if combined with a change in the injection rate. For example, as shown in Figure 4.4-1, a retardation of injection timing from 6 to 2 degrees before top center (BTC) produced a 32% reduction in NO_x, but this also reduced fuel economy by 10% and increased HC and CO emissions by 35% and 20%, respectively. Similar results have been obtained with other engines (other than Opel) and it is generally concluded that NO_x decreases significantly with fuel injection retardation, but fuel consumption increases markedly, as do HC and CO emissions. Also, smoke tends to increase as injection is retarded, depending upon design.

Changes in injection rate can be brought about by either changing the nozzle orifices and/or pump plunger displacement characteristics. In direct injection engines the fuel injection rate has been found to have a strong effect on NO_x formation. In the case shown in Figure 4.4-2, a 20% reduction in NO_x was obtained in the high load region of the engine by increasing the rate of injection from 5.7 to 8.3 mm^3/deg. and this proved to be fairly independent of timing. Pre-chamber engines are generally considered to be less sensitive to fuel injection rate, but this does not seem to be true where NO_x is concerned. For the case shown in Figure 4.4-3, the introduction of a larger diameter plunger (from 9 to 11 mm) resulted in a 20 to 35% increase in NO_x over the steady state load range of the engine. Changes in the injection rate brought about by changing nozzle orifice

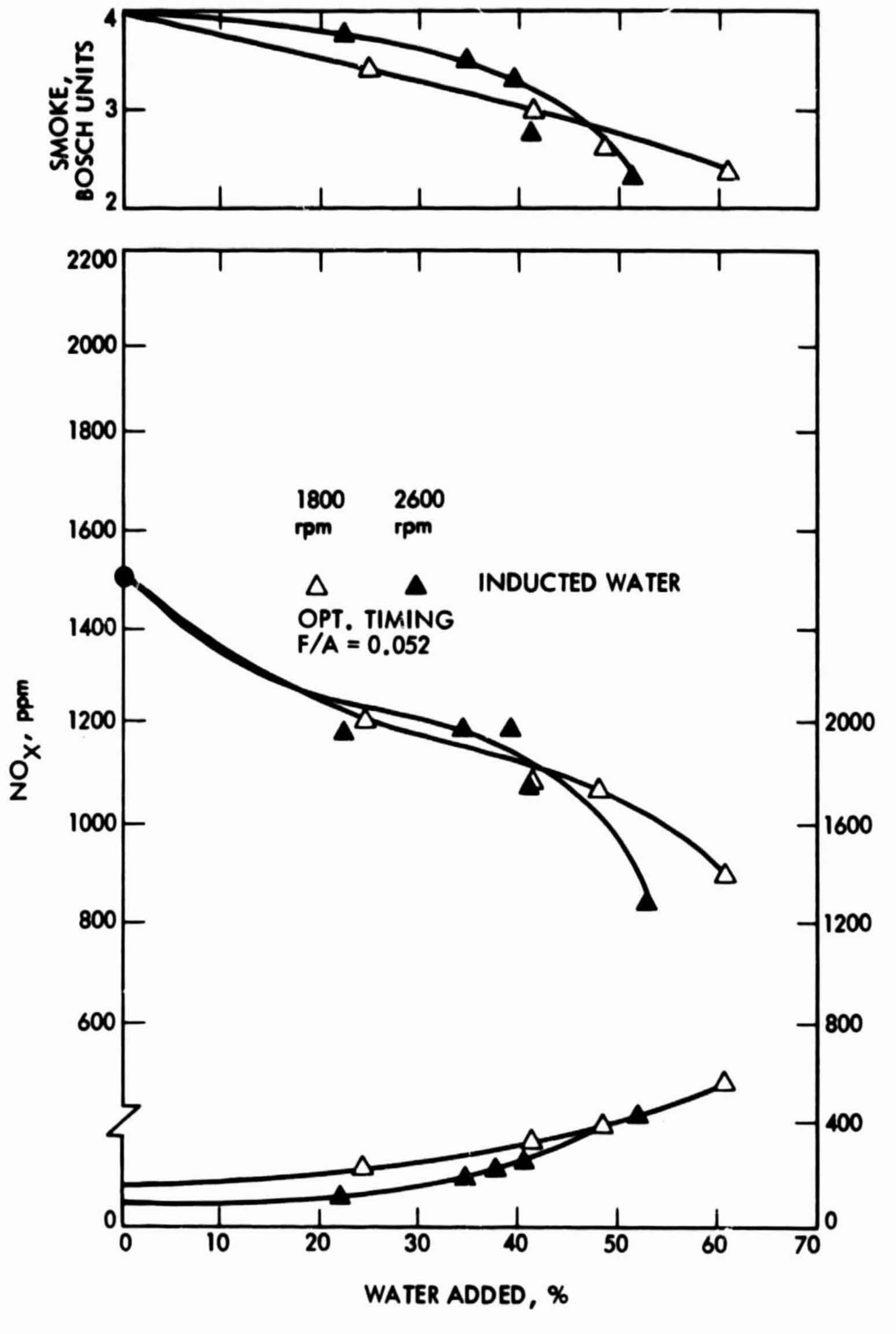


Figure 4.3-2. Effect of Inducted and Emulsified Water on the HC, NO_x and Smoke Emissions of an Open Chamber Diesel Engine (Ref. 32)

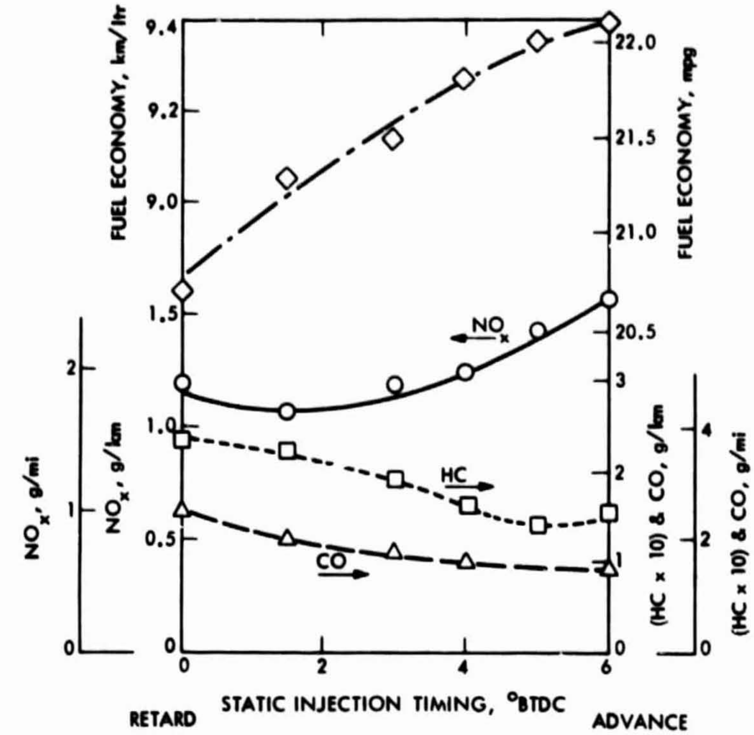


Figure 4.4-1. Effect of Injection Timing on Emissions and Fuel Economy of 2.1-ℓ Opel Diesel Engine (Ref. 32)

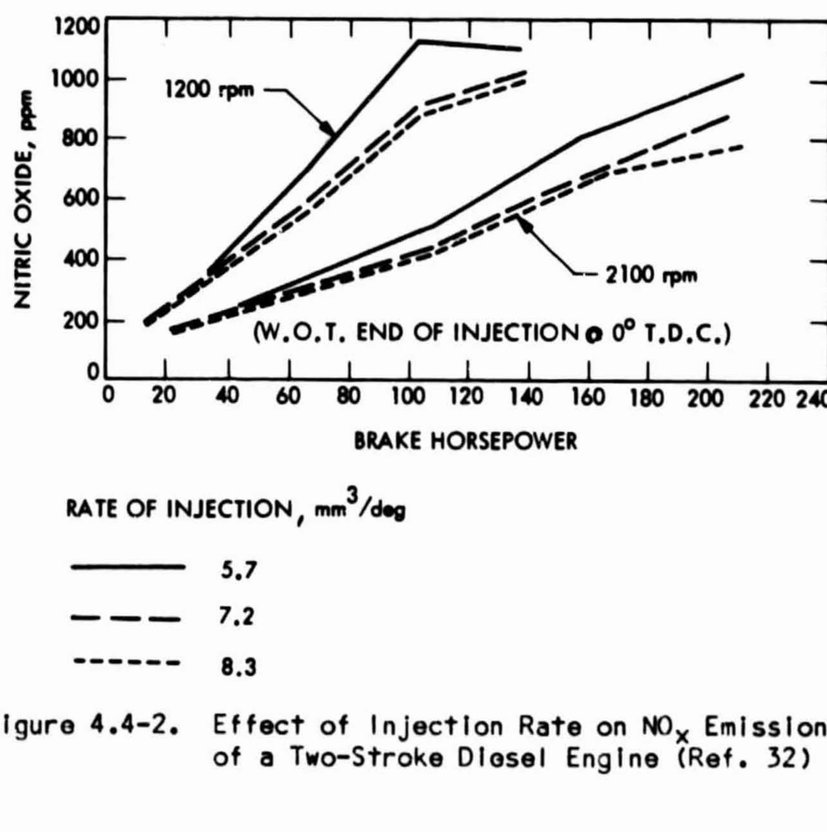


Figure 4.4-2. Effect of Injection Rate on NO_x Emissions of a Two-Stroke Diesel Engine (Ref. 32)

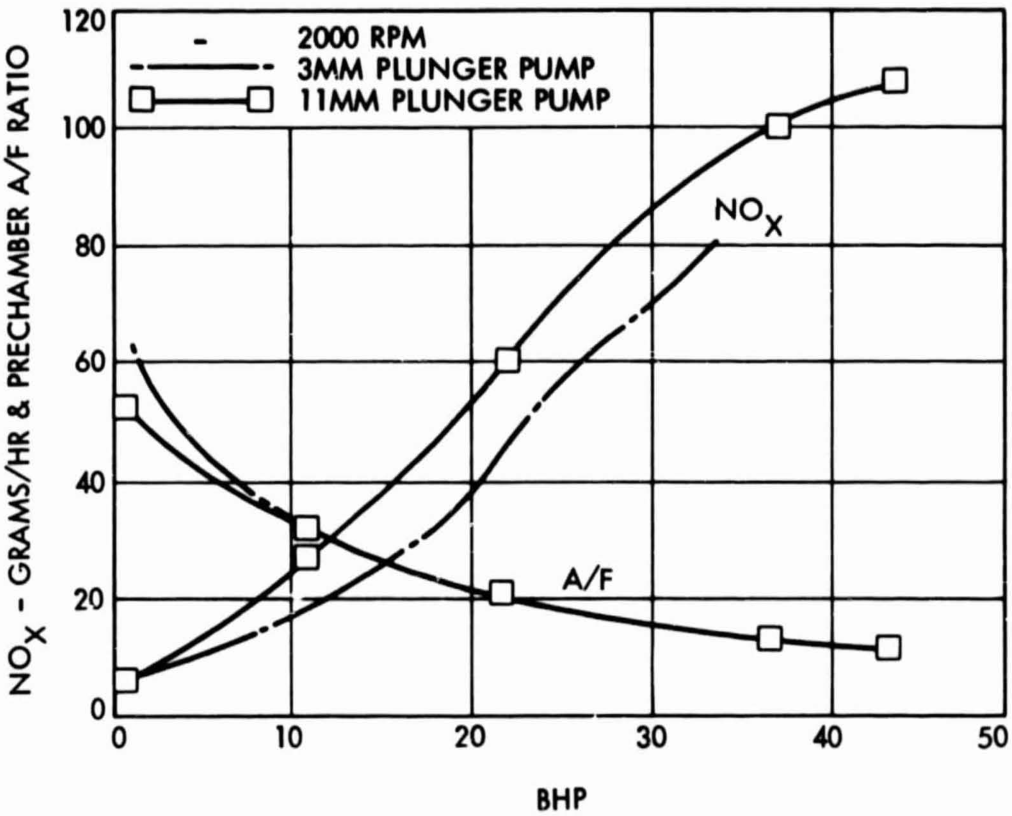


Figure 4.4-3. Effect of Injection Rate on NO_x Emissions (Ref. 32)

size and/or number have a compounding effect on NO_x development and combustion efficiency, because both the injection rate and spray pattern are changed (Figures 4.4-4 and 4.4-5).

The relationship between all the involved parameters is very complex and it is difficult to generalize the results. In one typical case, a 20 to 40% reduction in NO_x was obtained by enlarging the nozzle orifice diameter from .0055 to .0065 inches without changing the number of orifices. However, this improvement was associated with increased fuel consumption (3%), CO (70 - 100%) and smoke levels, indicating that the reduction in NO_x was obtained at the expense of combustion efficiency. Although data pertaining to the effects of injection orifice and spray angle on emissions are relatively scarce some benefits can be realized through empirical system optimization for a given engine design. The ultimate goal for the future is an electronically controlled and programmed system that allows for independent variation of all variables involved.

According to press information (Ref. 33), General Motors takes a new approach to more accurately timed diesel combustion to reduce the particulates. As shown in Figure 4.4-6, the symmetrical pattern of microwaves reflected by the piston upon approaching and passing the top dead center are used to accurately determine the location of the true dead center. In addition, a luminosity signal generated from combustion is then used to adjust injection pump timing relative to the dead center. In the currently proposed version, the microwave probe is inserted instead of the existing glow plug for

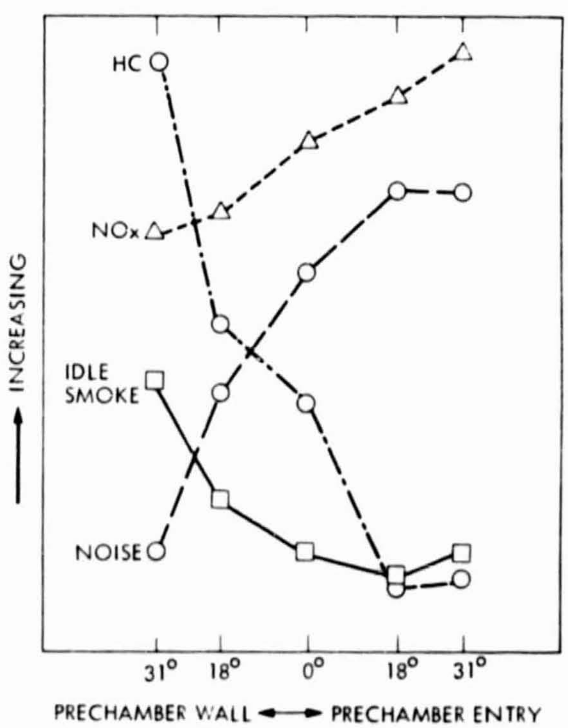


Figure 4.4-4. Effect of Nozzle Hole Angle (Ref. 28)

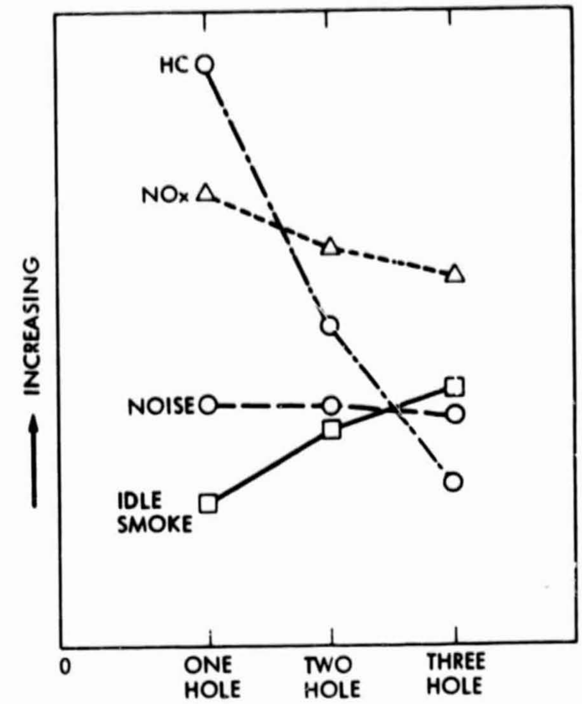


Figure 4.4-5. Effect of Multiple Holes (Constant Flow Area) (Ref. 28)

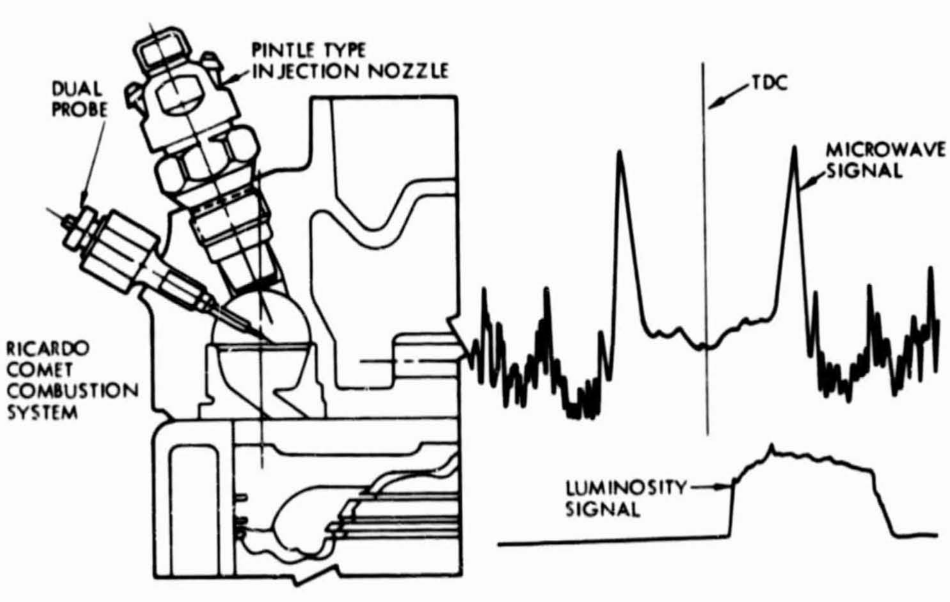


Figure 4.4-6. General Motors Diesel Microwave Timing Method (Ref. 33)

timing checks and adjustments in trucks. The probe will then be removed and the glow plug inserted for engine operation. This method has the potential for a continuous scanning and measuring of piston position and combustion intensity to variably control injection rates according to optimized schedules.

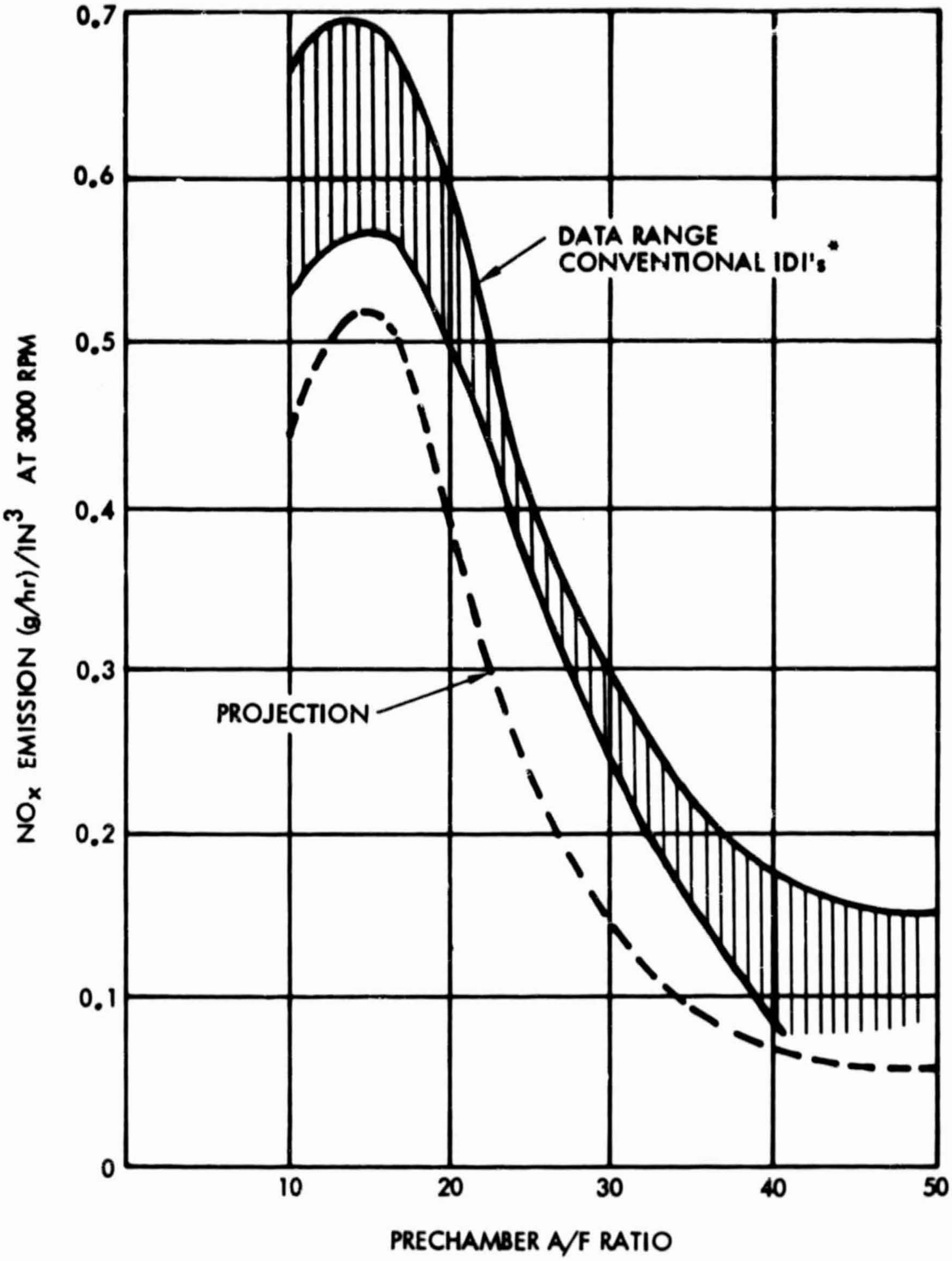
4.5 COMBUSTION CHAMBER DESIGN

Shape and volume of the pre-chamber have a strong effect on the air-to-fuel ratio and internal flow velocities, which in turn cause changes in combustion efficiency, and NO_x emission in particular.

For a given pre-chamber, increases in the air-to-fuel ratio have been shown to produce a drastic reduction in NO_x. For example, as shown in Figure 4.5-1, increasing the A/F ratio from 25 to 35 resulted in NO_x reduction by a factor of more than two. If brought about by configurational changes of the chamber, the effects of A/F ratio are more difficult to define because factors other than air motion and spray pattern are involved. In pre-chamber engines, increased swirl intensity has improved combustion efficiency but, as shown in Figure 4.5-2, has also resulted in an unfavorable NO_x-particulate trade-off. For a given smoke density, NO_x emissions were found to be lowest with turbocharged engines. Improved mixing and increased reaction rates resulting in higher combustion temperatures are primarily responsible for this. The ratio of the pre-chamber volume to cylinder volume, the throat dimensions in relation to the chamber, and the shape of the chamber itself have also been shown to have certain effects on NO_x and HC emissions. However, the magnitudes and trade-offs, especially between NO_x and fuel efficiency vary from one design to another and require individual treatment for optimization.

Figure 4.5-3 for example, shows three combustion chamber designs that were under consideration for the Olds diesel engine described in Section 3.5. The configurations differ primarily in the relationship of the position of the chamber and the injection nozzle to each other. In design "A", fuel is injected tangentially with the swirl on the inboard side of the chamber. Design "B" directs fuel across the chamber inboard of the chamber center, and design "C" directs the fuel through the chamber center. There are also differences in throat design. In "A", the throat is aimed towards the center of the pre-chamber, changing gradually in "B" and "C" to a direction nearly tangential to the chamber wall, i.e., design "A" actually represents an angled off pre-chamber, design "C" a swirl chamber, and design "B" is in between.

As mentioned in Section 3.5, these chambers were already selected from a large variety of configurations as the most favorable designs. In the final heavily emission-oriented evaluation, design "A" was chosen for further development by Oldsmobile. This design produced the lowest NO_x emission and the lowest noise, but was highest in HC. Ways of further reducing HC without increasing NO_x were then evaluated with design "A" by varying the nozzle spray angle and the number of orifices. This produced the trends shown in Figures 4.4-4 and 4.4-5. According to GM (Ref. 28) HC could be reduced by 70% with this approach without significantly increasing noise or NO_x, but smoke would be increased to a visible level during engine idling.



*Indirect injection

Figure 4.5-1. NO_x Emissions versus Pre-Chamber Air-Fuel Ratio (Ref. 1)

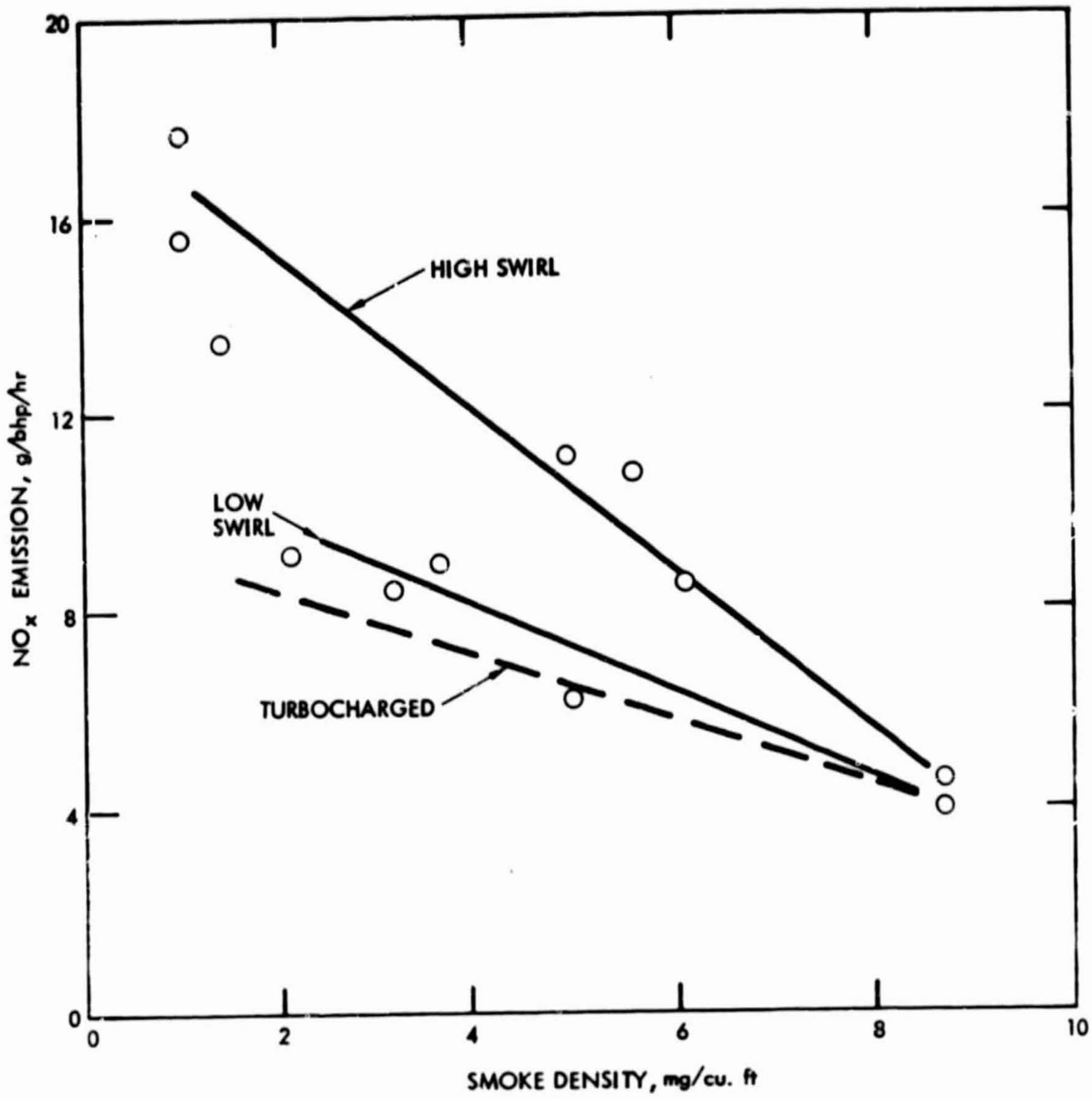


Figure 4.5-2. Effect of Air Swirl and Turbocharging on NO_x Emission of Open Chamber Diesel Engines (Ref. 32)

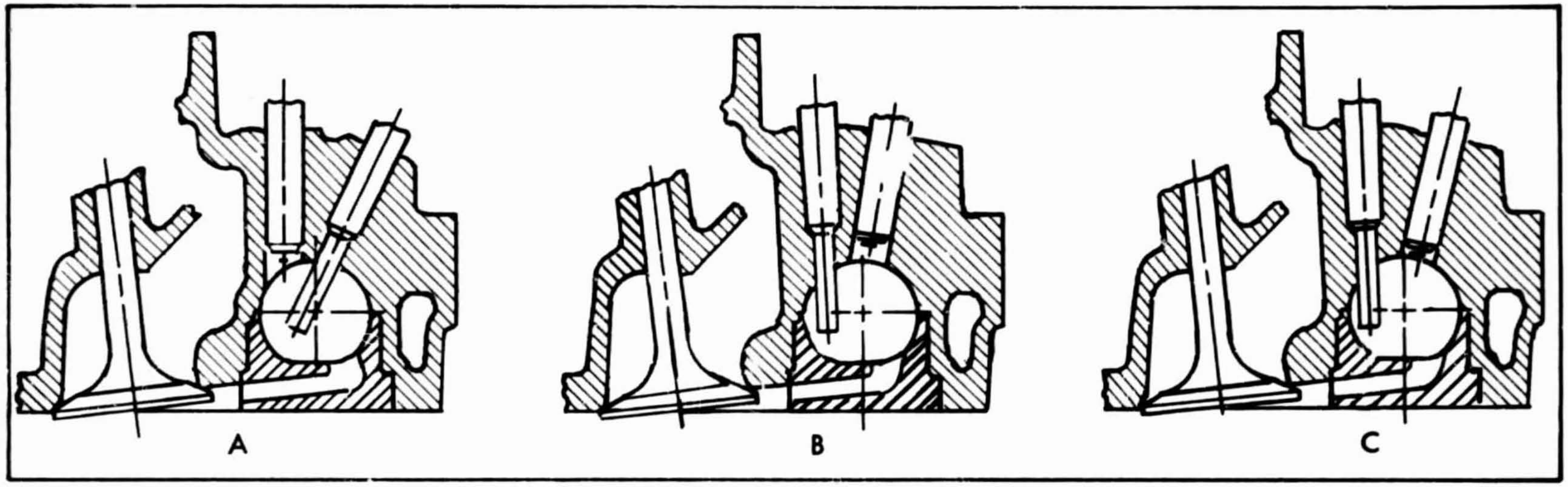


Figure 4.5-3. Chamber Configurations Considered for Final Design of Oldsmobile Diesel Engine (Ref. 28)

A = Selected Design
B & C = Alternate Designs Tested

This example clearly shows the difficulty involved in making clearcut recommendations for combustion chamber improvements. Unless one can realistically model the relationship of all factors involved, the engine designer must depend largely upon his own engineering judgement, instinct, and a certain amount of luck to detect and to feel his way through the improvement potential.

In general, the naturally aspirated divided chamber engine appears to have a small potential for improvement in the existing HC-NO_x relationship. The naturally aspirated engine depends on high intensity air motion for adequate mixing at low A/F ratios, which are needed to produce an acceptable power concentration. Severe penalties are encountered in combustion efficiency, and in smoke and particulate emissions, if lower swirl intensity and higher A/F ratios are used to achieve a reduction in NO_x emissions. Developers are therefore very reluctant to tamper with a successful chamber design.

Turbocharged divided chamber engines which generally have been shown to be lower in NO_x also benefit from the A/F-swirl-NO_x relationship. They generally operate at relatively high A/F ratios because of thermal and structural limitations. Turbocharged engines are also less dependent on swirl for mixing, because A/F ratio, compression end temperature and general flow turbulence are higher than in naturally aspirated engines. They have been found to be relatively insensitive to further refinements in existing pre-chamber designs.

R&D approaches are being taken in the direction of injection open chambers, and evaporative combustion. These will be discussed under "Efficiency Improvement Potential" in Section 5.5.

4.6 EXHAUST GAS RECIRCULATION (EGR)

Most diesel experts currently feel that a NO_x level on the order of 1.5 g/mi can be achieved by means of the modifications and refinements described above. Compliance with a 1.0 g/mi standard for NO_x will require the introduction of EGR, i.e., the dilution of engine intake air with exhaust gases. This results in lower combustion temperatures and reduced oxygen concentration, which together, tend to inhibit the formation of NO_x.

As illustrated in Figure 4.6-1, EGR is an effective and fairly predictable means of reducing the formation of NO_x, but its use is associated with severe penalties in engine efficiency, smoke and the other controlled pollutants HC and CO. These penalties vary from engine to engine, but become unacceptable beyond a certain percentage of EGR. The increase in HC and smoke (Figure 4.6-2), which is attributed to reduced oxygen during combustion with EGR, becomes particularly pronounced in the low speed high load region of the engine, and the EGR percentage must be reduced in these regions to avoid unacceptably high levels of smoke and particulate emissions.

The implementation of EGR in diesels requires a valve-controlled modulated recirculation system similar to those used on gasoline engines. However, the presence of relatively large amounts of particulate matter in diesel combustion products presents system contamination problems that require

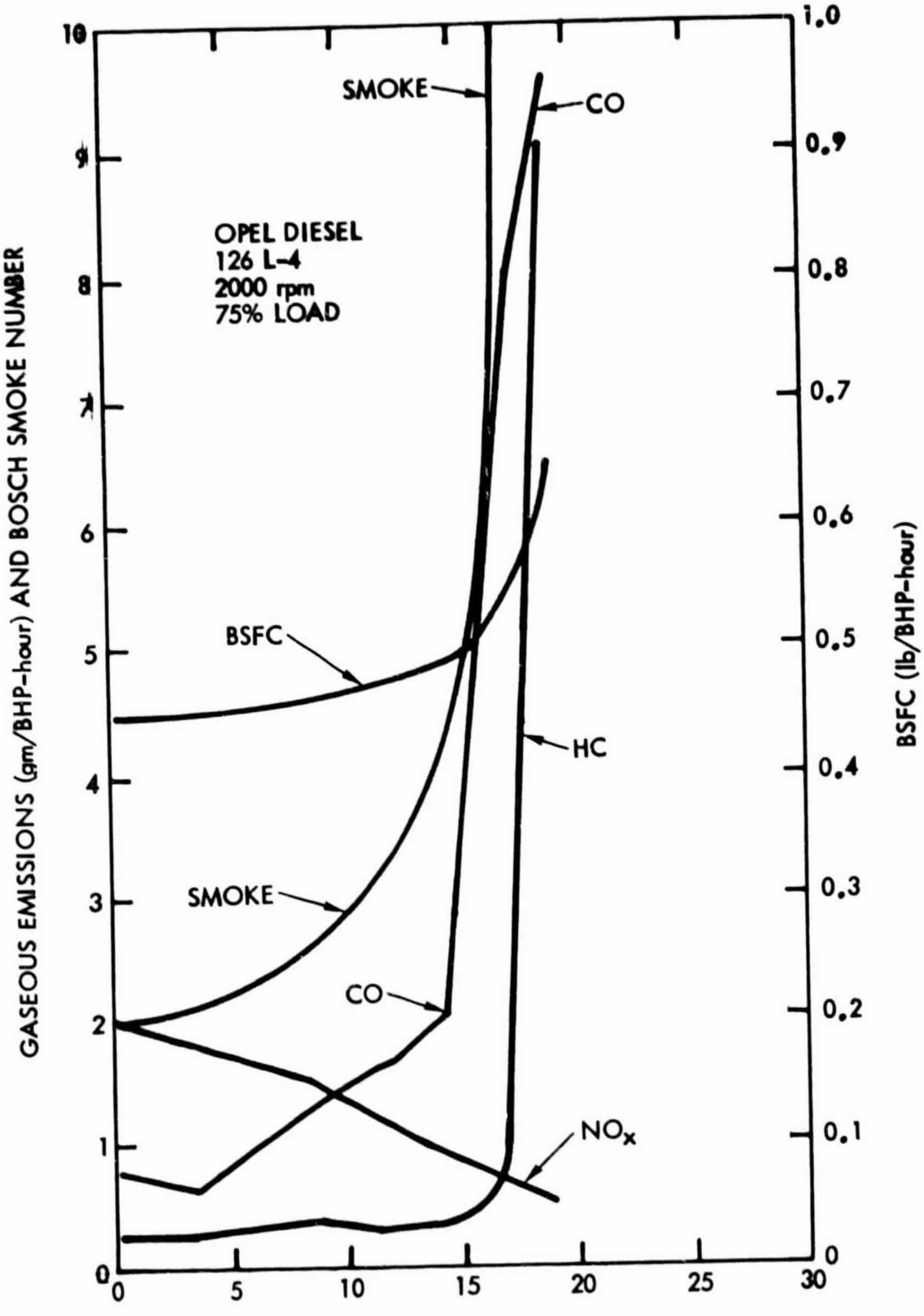


Figure 4.6-1. Emissions vs. EGR Flow Rate (Ref. 32)

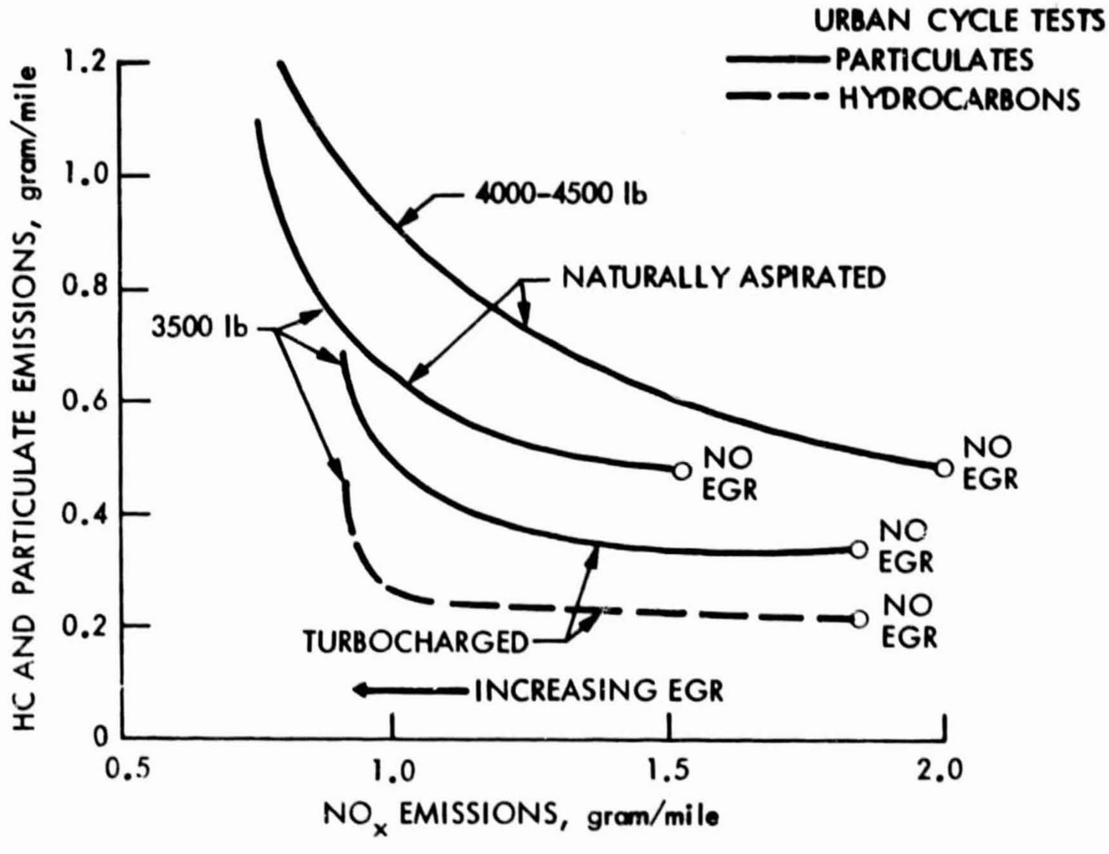


Figure 4.6-2. Effect of EGR on NO_x, HC and Particulate Emissions (Ref. 34)

special design and development approaches to make diesel EGR systems reliable and serviceable. As in spark ignition engines, EGR contaminates the engine oil, requiring reduction of time between oil changes, which is already less than that required for gasoline engines. Mercedes Benz has reportedly developed a new crankcase system that inhibits oil contamination with carbon particulates and raises the oil change interval in conventional (non-EGR) engines from 3000 to 5000 mi (Ref. 17). This would ease the oil contamination problem with EGR to a certain degree. Reportedly Olds will adapt the same system to the 1981 model engines.

Internal EGR can be induced to a certain degree by variation of valve timing and overlap by lowering cylinder scavenging efficiency. However, an optimization of the EGR percentage throughout the engine operating range will require the addition of complex mechanical systems to achieve variable valve timing. To assure sufficient valve-piston clearance, the relatively large valve overlap required would also necessitate the use of piston designs that could have an adverse effect on combustion efficiency.

In addition to the discussed problems, concern has also been voiced (Ref. 35) regarding other potential side effects with EGR that must be taken into account. While NO_x is produced at peak cycle temperatures, it may combine with excess oxygen during expansion at lower temperatures to form NO_2 (nitrogen dioxide) which is not a regulated pollutant. NO_2 can combine in sunlight to form other substances, including nitric acid, which can have inhibiting effects on plant life.

In summary, EGR is an effective and predictable method of reducing NO_x, but associated side effects and system reliability problems make its application questionable from the standpoint of economy. Contrary to widespread belief, it does not seem to be the answer to the emission problem.

4.7 EXHAUST AFTER TREATMENT

Currently known reduction catalysts used in gasoline powered cars are not effective in controlling diesel NO_x because of the highly oxidizing nature of diesel exhaust. Although catalysts can be effective in reducing HC and CO emissions, the deposition of soot on the catalyst surface and a rapid degradation caused by poisoning of the active material are still unresolved problems. Thermal reactors, which are effective in reducing HC and CO emissions in the exhaust of spark ignition engines, are not very effective in diesels. Diesel HC and CO concentrations are generally too low to sustain oxidation, and the pollutants of major concern, NO_x, smoke and particulates, are not substantially reduced.

Besides NO_x, a major problem of concern is that of particulates. Because of the difficulties in solving the particulate problem from the combustion side, the entrapment of particulate matter in the exhaust is strongly being considered. Entrapment is not the most desirable approach, but it is straight forward, easier to understand, and may well be the only way out of the dilemma until the physical causes leading to the formation of particulates are better understood, and more effective measures can be devised. The

undesirable aspects associated with entrapment are the need for an additional system that must be serviced. If feasible, it cleans up the exhaust but does not alleviate engine oil contamination problems. The amount of particulate matter generated by automotive diesels ranges between 0.3 and 1.0 g/mi, compared to 0.3 g/mi for uncontrolled, and 0.01 g/mi for catalyst-controlled spark-ignition engines cars using unleaded gasoline.

The major problems with diesel exhaust filtration are the physical characteristics of diesel particulates and their size. More than 65% of diesel exhaust particulates are $\leq 1\ \mu m$, and 50% are $\leq 1/2\ \mu m$ in size, which makes a continuous removal by means of vortex filters such as cyclones very inefficient. Filtration devices, such as steel wool, stacks of wire mesh, and spiral wound filters have proven fairly efficient, but clogging is a major problem. Because of the fluffy and sticky nature of diesel particulates, the devices become plugged after a few hours of operation resulting in excessive performance losses because of increased exhaust back pressure.

A periodic burning of collected particulates is widely believed to be the answer to the clogging problem. Unfortunately, diesel exhaust temperatures usually do not reach the 900 to 1000 °F (480-540 °C) temperature needed to ignite and to incinerate soot particulates, which mainly consist of carbon and small amounts of unburned hydrocarbons. Alternatives for particle burnoff include periodically installing spark ignited burners using diesel fuel, periodic throttling of the engine, and temporary operation on a fuel rich mixture to raise the exhaust temperature to the above indicated level, as well as electrical methods such as sparks and incandescent wire mesh filters and, perhaps, lasers. In any case, back pressure will probably be used to trigger the burn-off whenever a critical exhaust pressure level has been reached.

A unique approach to the burn-off problem is currently taken by General Motors using the dual trap concept shown in Figure 4.7-1 (Ref. 8). The system consists of two filters in parallel having separate electrical heating elements and a flapper valve that routes only a small fraction of the exhaust flow to the filter that is being heated, as necessary to provide enough oxygen for the burn-off. Most of the exhaust then flows through the filter that is not being heated. The system is estimated to use about 500 W during burn-off periods.

Based upon work already done (Ref. 36), Corning Glass Works has come up with a ceramic filter substrate that, according to tests, effectively entraps 80% of diesel exhaust particulates. The filter material is a silica, magnesium, and alumina composition that can withstand continuous burn-off of carbon particulates under controlled conditions, and is claimed to remain efficient for 50,000 mi before a filter change is necessary.

To avoid burn-offs and filter changes, TRW Inc. (Ref. 37), takes the approach to a conventional pleated bag-filter miniaturized for diesel exhaust flow rates. The cake of particles collected on the filter cloth is periodically removed by mechanical shaking. The device proposed is schematically shown in Figure 4.7-2. TRW suggests using a fiberglass cloth pleated

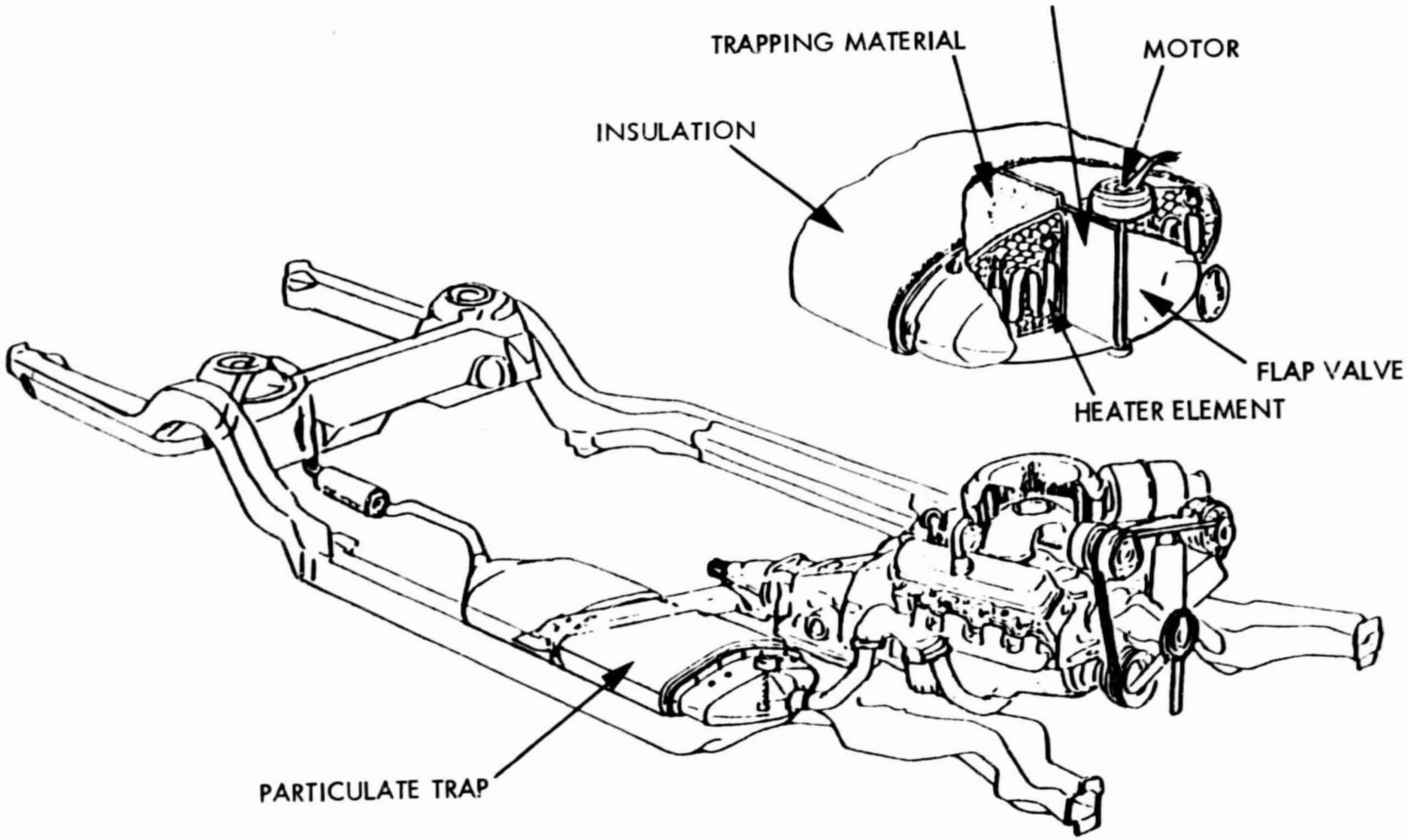


Figure 4.7-1. Dual Path Trap/Incineration Device Proposed by General Motors (Ref. 8)

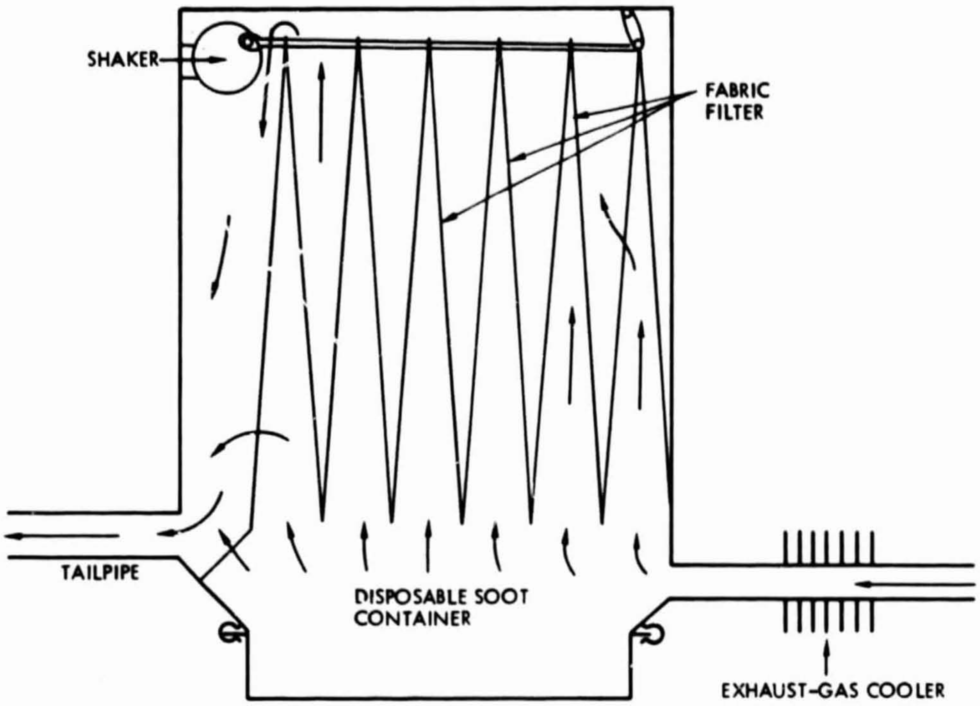


Figure 4.7-2. Schematic of Mini-baghouse for Filtration of Diesel Exhaust Proposed by TRW, Inc. (Ref. 37)

over a wire frame and mounted into a rectangular box that will fit in the crush space behind the gas tank. The filter cloth will have to be shaken every 500 - 1000 mi to keep the back pressure within acceptable limits.

According to industrial experience, it appears doubtful whether a low back pressure can be repeatedly restored with filter shaking alone. The in-depth penetration and clogging characteristics peculiar for diesel soot will probably require more effective filter cleaning methods, such as reverse pulse flow, for example. Effective pressure pulsing by using compressed air available at filling stations also appears to be more reliable and more cost effective than the shaking mechanism shown in Figure 4.7-2. A variety of researchers have pursued the approach of coagulating diesel particulates into larger sizes that allow for a continuous separation from the flow by using straight through or cyclone type vortex seperators.

Under a program sponsored by the EPA, the University of Arizona has experimented with an electrostatic filtering device which has shown encouraging results (Ref. 38). The proposed device (Figure 4.7-3) consists of a corona wire extending through the axis of a cylindrical pipe that is negatively charged with -25 kV. A simple clog free swirler is provided upstream to induce vortex flow which creates radially outward displacement of the particles captured by the electrical field. According to Figure 4.7-4, a noticeable removal capability has been demonstrated for all sizes below 6 μm. The problem of collection and retention of the particles removed from the flow was not addressed in this program. Figure 4.7-5 shows a similar device tested by General Motors.

Two other prospective concepts for continuous diesel exhaust clean up are under consideration at the Jet Propulsion Laboratory. One, referred to as a "Spark Destructor System," uses an electrically charged grid structure similar to the ones used on the bug-eliminators currently marketed in the United States. The device also functions in a similar way. Diesel particulates passing through the grid plane are agglomerated upon approach and induce sparking when passing between the grid elements which brings about a burn-off of the agglomerates. According to initial experimentation, this seems to be a workable approach that should be pursued further.

The other concept under consideration, as schematically shown in Figure 4.7-6, consists of a vortexing agglomeration device (similar to the one used by General Motors) in series with a cyclone, which is more size-effective than a flow-through vortex filter alone. EGR flow is exhausted from the lower end of the cyclone together with separated matter that will collect in a disposable bag. The extraction of flow from the lower end of the cyclone greatly improves cyclone efficiency with smaller particle sizes. For maximum efficiency, the cyclone will be sized for low engine speed where most particulates are generated, and can be bypassed at higher engine speeds from the clean core of the agglomerator as necessary to keep the back pressure within acceptable limits. It is expected that 99% of all particulate mass greater than 2 μm, and greater than or equal to 80% of the total mass can be continuously removed by the proposed system which is strongly dependant on cyclone speed and on the effectiveness of the agglomerator. Possible methods other than electrical fields under consideration for agglomeration are mechanical baffling, pneumatically induced standing shock waves and focused sonic irradiation.

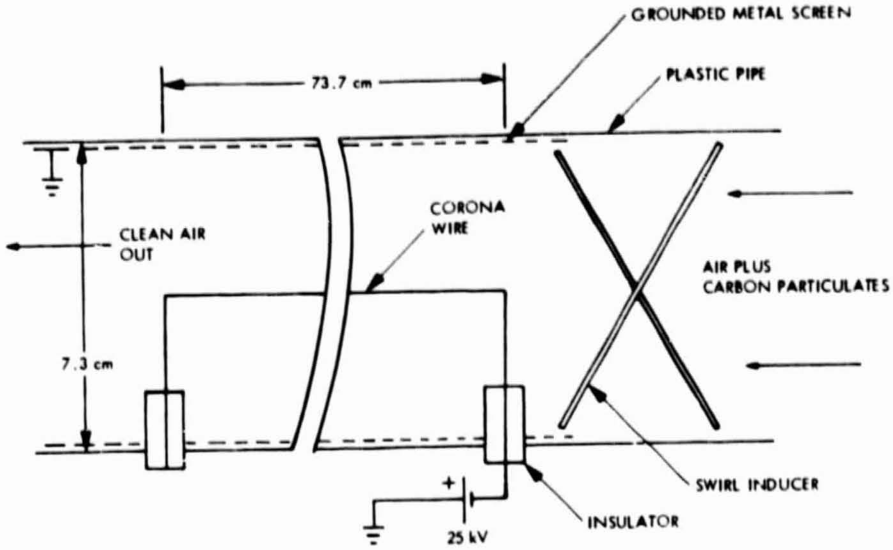


Figure 4.7-3. Schematic of Helical Flow Electrostatic Agglomerator and Separator Testing by the University of Arizona (Ref. 38)

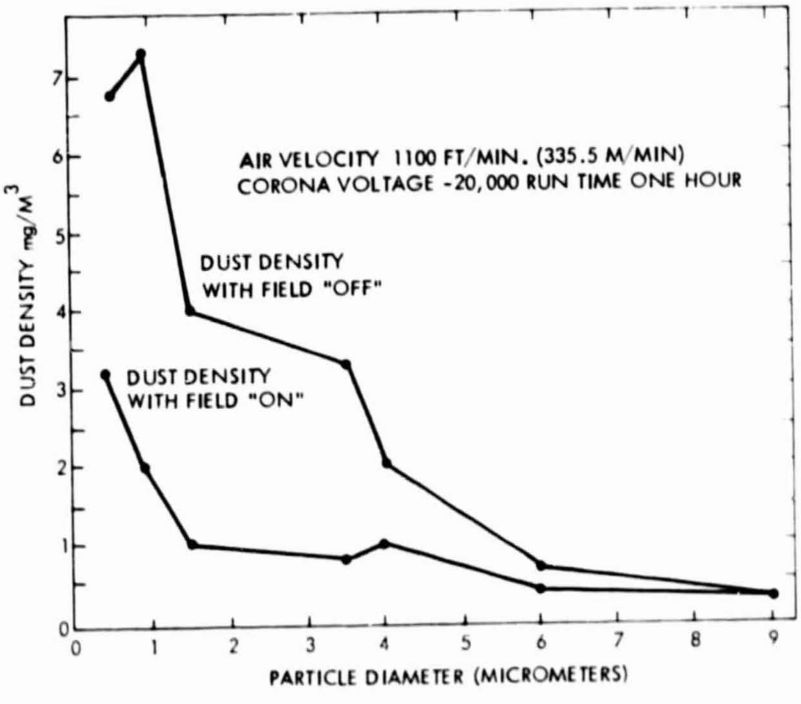


Figure 4.7-4. Demonstrated Removal Capability of Helical Flow Electrostatic Agglomerator (Lab Tests, Control of Simulated Diesel Smoke by Means of a Swirling Flow Electrostatic Precipitator) (Ref. 38)

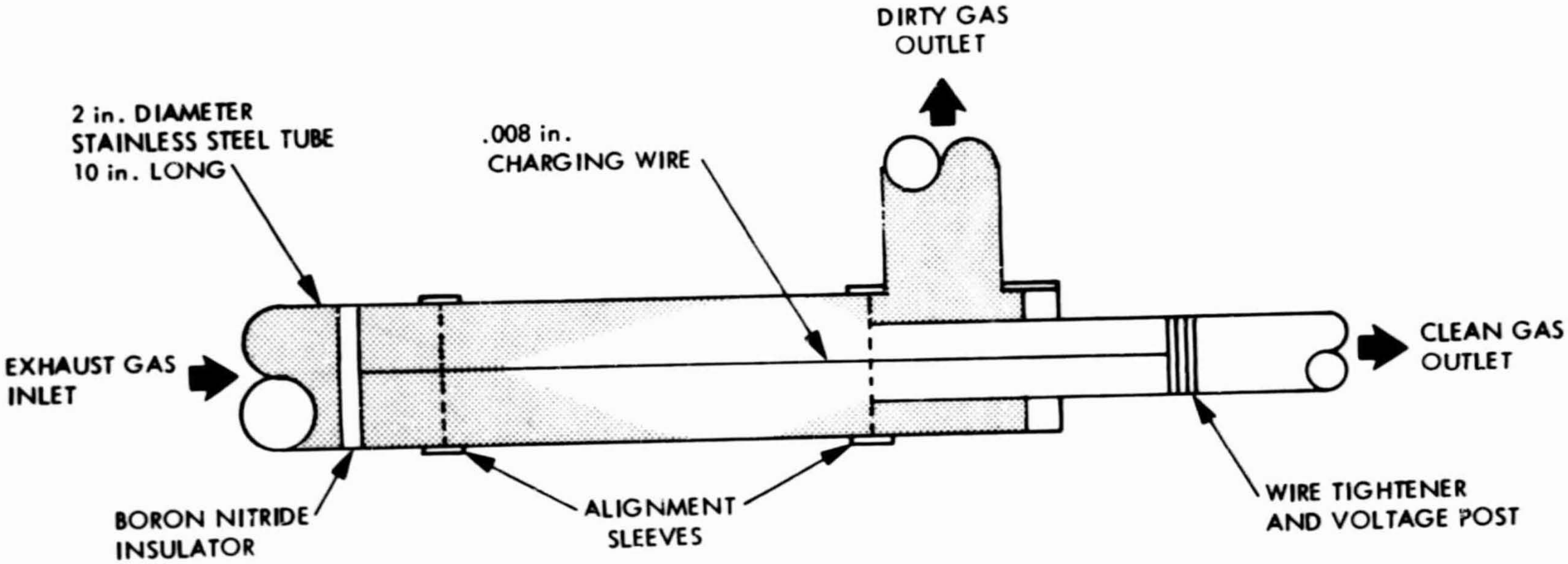


Figure 4.7-5. Schematic of Electrostatic Agglomeration Device Tested by General Motors (Ref. 8)

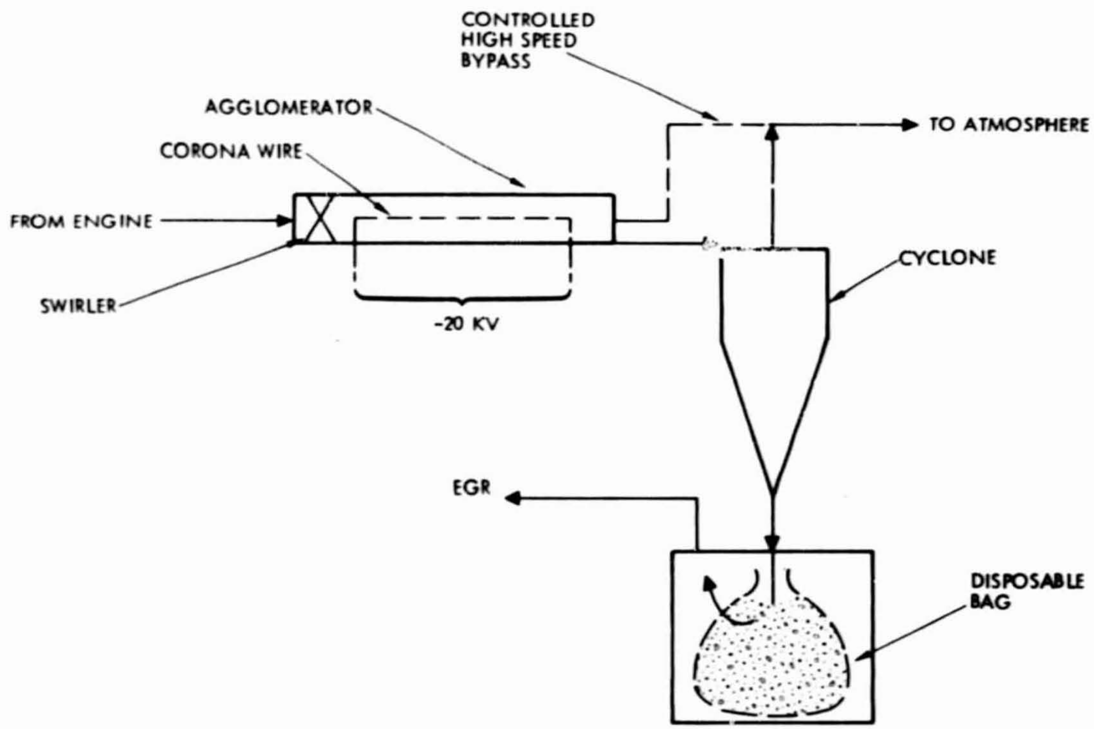


Figure 4.7-6. Design Concept for Diesel Exhaust Filtration Utiizing Electrical Agglomerator in Series with Bypassable Split Flow Cyclone and EGR Through Disposal System

All of the approaches taken or under consideration require some energy in the form of exhaust pressure losses, electrical energy generated from shaft horsepower, or extra fuel for particle burn-off. The 500 W for burn-off which General Motors has estimated for the Olds diesel trap, represent a power loss on the order of one half percent of the nominal power. Each inch of water of exhaust back pressure caused by the filter represents an additional power loss on the order of 0.1% of the parasitic losses already in existence. With a pressure drop on the order of 2.5 to 5.0 in. of water common for most filtration devices, this amounts to a power loss on the order of 0.5% relative to nominal engine brake horse power, depending upon clogging characteristics and cleaning intervals. Although these numbers represent a very rough estimate, it appears that approaches that do not require extra energy for particle burn-off should be most vigorously pursued, although periodic servicing will be required.

4.8 FUEL MODIFICATIONS AND ADDITIVES

Fuel properties such as those determined by composition, aromatic content, cetane number and boiling points of the constituents, have been found to have a small to moderate effect on particulates and organic compounds emitted from diesel engines. Existing data are not conclusive because fuel properties also affect exhaust emission changes by altering the engine operating characteristics, which are strongly design-dependent.

Indications are that lighter and more volatile fuels, such as Diesel No. 1, produce 10 to 40% less particulates than Diesel No. 2, depending on engine design, engine/vehicle configuration and drive cycle (Ref. 32, 34). Limited data show reductions of HC, CO and NO_x on the order of 10% as fuel cetane numbers are increased from 40 to 50.

Certain fuel additives containing barium (Ref. 32) have been found quite effective in reducing carbon particles and smoke by 35 to 60%, with effectiveness increasing with increased barium content. Seventy percent of the barium is harmlessly exhausted from the engine in the form of barium sulfate, but the remainder consists of soluble compounds, some of which are toxic. Engine manufacturers do not recommend the use of additives containing barium because of the adverse effect they have on the durability of the engine and its fuel injection system.

SECTION 5

POTENTIAL FOR FURTHER IMPROVEMENT

5.1 BACKGROUND AND SCOPE

Further improvement in diesel performance and economy must be achieved to sustain the economic advantages of the diesel. There does not seem to be an easy way of meeting future NO_x and particulate emission requirements without impairing diesel performance and fuel economy. This makes the future of the automotive diesel questionable from the consumer and general energy standpoint, particularly when the improvement potential of the gasoline engine is taken into consideration (Table 2.2-4).

The diesel still has a great potential for improvement in many areas. The major contributors to diesel and gasoline engine losses are heat losses through the engine cooling system, and heat leaving the engine through the exhaust. As shown in Figure 5.1-1, the diesel total heat loss accounts for almost 2/3 of the total energy input to the engine. The remaining loss of 2.7% are primarily because of speed dependent mechanical, parasitic and internal pumping losses (Table 5.1-1). Although relatively small in terms of energy, each of these contributors still has potential for improvement in terms of BMEP.

Losses because of incomplete combustion are relatively small in modern diesel engines, but further research and improvements in this area are vital from the standpoint of emission reduction. Many of the programs in progress are, therefore, primarily related to emissions and health effects, and some are directed toward improvements that will reduce the negative by-products of emission-reducing measures such as EGR. Some effort is being made to penetrate the large potentials for improvement in such areas as engine insulation using ceramics, and to improve the fundamental understanding of what occurs inside the diesel engine combustion chamber. The following sections present selected approaches for performance and efficiency improvement that appear to be most promising and effective at the present time. The trend shown in Figure 5.1-2 is believed possible if the potential for improvement is fully used.

5.2 TURBOCHARGING

Turbocharging has successfully penetrated the small high speed diesel field, including the 1.5-ℓ (VW Rabbit) class, and can be considered an economically feasible solution to a number of diesel problems, particularly those of driveability.

Because the efficiency of present turbocharger systems is already high, a further increase in engine efficiency with turbocharging will be very small. Changes will come very slowly, and will likely be aimed at improving hardware design and production technology, thereby making turbochargers more cost effective and more responsive. Also, new technology must be

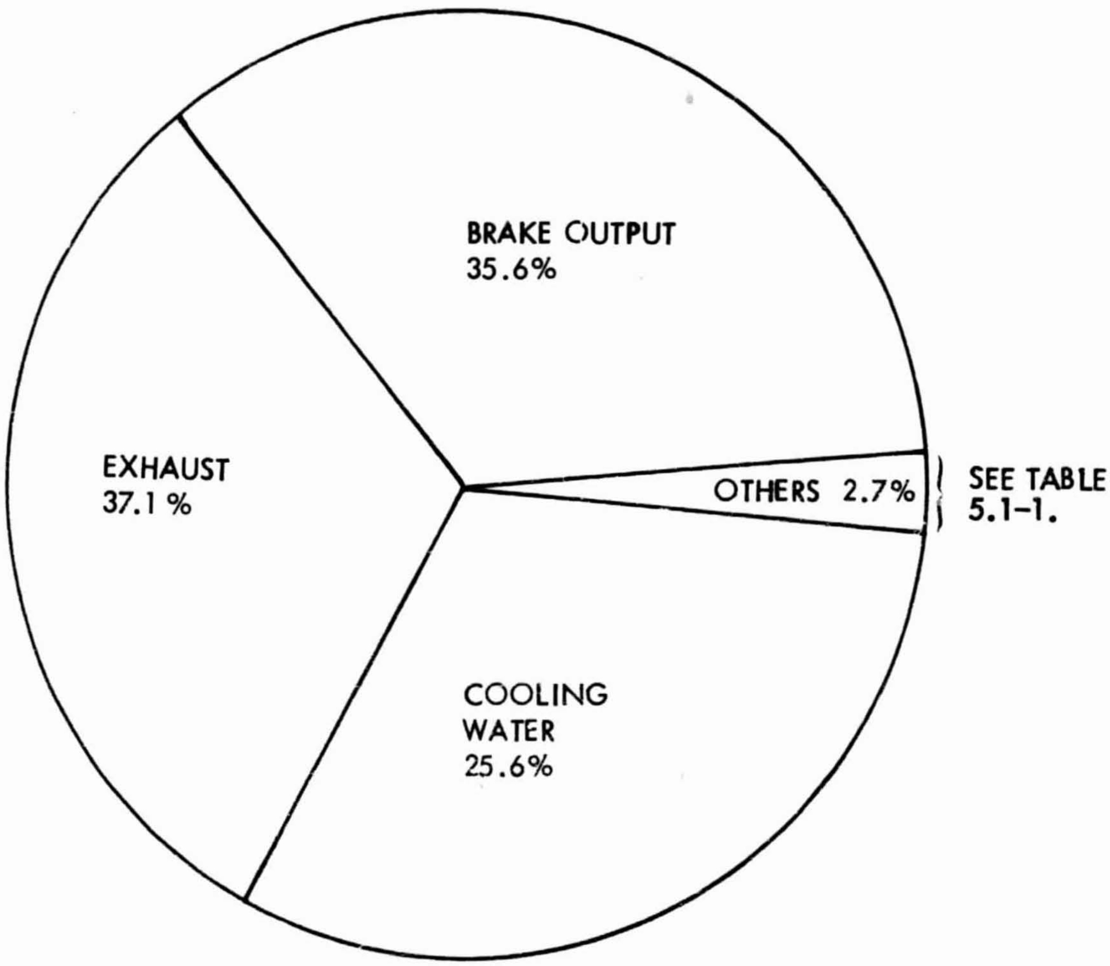


Figure 5.1-1. Typical Energy Balance of Diesel Engine (Ref. 39)

Table 5.1-1. Typical Breakdown of Mechanical Losses in Diesel Engines (Ref. 5)

	%
Friction Piston	40 - 45
Friction Bearings	25 - 30
Valve Train & Gears	14 - 17
Water + Oil Pumps	9 - 10
Injection Pump	6 - 8
	100%

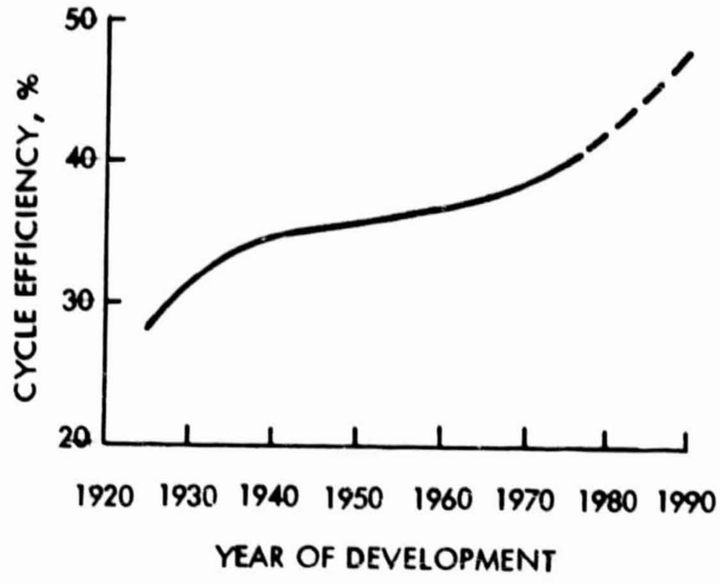


Figure 5.1-2. Diesel Cycle Efficiency Trend (Ref. 40)

sought in adapting turbochargers to smaller engines. Major problems to be overcome in this area are small size effects such as Reynolds number effects, surface roughness, rotor clearance, general aerodynamics quality, bearing design and rotor stresses.

Certain improvements in efficiency will result from overall systems optimization, including inlet and exhaust tuning. For option efficiency, future turbocharged engines will not be derived from existing naturally aspirated engines, but will be designed and developed from the start as a turbocharged system.

5.3 INTERCOOLING

Intercooling, or aftercooling, is feasible with present technology. Its application is mainly a matter of cost, packaging, reliability and maintenance. Air coolers have a limited improvement potential with regard to size and pressure drop. Considerable improvement in weight may be possible through the introduction of advanced materials and production methods (thinner walls). The proper integration of an intercooler into the engine may possible be the most attractive and economically feasible solution for reducing inlet air temperature in automobiles.

Intercooling has a noticeable effect on NO_x emission (Figure 4.3-1) and volumetric efficiency, but at today's feasible boost pressure (approximately 0.7 bar), the penalties of increased weight, bulk and cost tend to negate its advantages. Intercooling will be more of an incentive in conjunction with future low-compression diesel engines (Section 5.7) that allow for higher inlet pressure than are practicable currently.

5.4 TURBOCOMPOUNDING

Turbocompounding involves the extraction of mechanical energy from the exhaust in excess of what is needed to drive the compressor. This excess energy is then fed back into the power take-off of the engine, or is used otherwise. The fuel economy gains that can be realized from the turbocompounding of water cooled, turbocharged diesel engines are on the order of 5%, and do not seem to warrant the effort except when high boost pressure ratios are structurally acceptable.

Turbocompounding will be a necessity to take advantage of reduced cooling losses, which will otherwise result only in higher exhaust temperature with no increase in efficiency (Section 5.8). For a fully insulated (adiabatic) engine, turbocompounding will result in an efficiency gain on the order of 25 to 30% over turbocharging alone, excluding expansion and transfer losses.

The conventional design approach for turbocompounding usually involves a separate power turbine downstream of the turbocharger, with useful work transformed by a speed reduction gear to the engine power take-off (Figure 5.4-1). Such a design does not seem to be too practical for a small automotive diesel, because of cost, bulk and complexity.

A prospective concept shown in Figure 5.4-2 uses a high speed alternator as an integral part of the turbine rotor to avoid gearing problems. Today's radial turbines are capable of handling expansion ratios that would make an extra power turbine unnecessary. In this case, turbocompounding does not involve greater mechanical complexity than is required for turbocharging alone. The energy generated can then be used to drive engine accessories such as fans and air conditioning, and to charge the battery. In the case of an insulated (adiabatic) engine, an excess of electrical power would be available that could be used either to power extra accessories or working equipment, or to feed an energy storage system.

Approaches in this direction are currently being taken by developers of diesel-powered heavy duty working equipment with a high demand for accessory power, such as Caterpillar, and John Deere. In conjunction with electrical or mechanical energy storage systems such as batteries or flywheels, electrical turbocompounding seems to have potential for use in medium to large-sized diesel-powered passenger cars. This should be further evaluated.

A hard look is also being taken at displacement-type, low-speed expanders that can be coupled to the engine by using a conventional chain or V-belt and pulley drive mechanisms. Candidate expanders concepts under consideration, primarily for use in high temperature steam and in organic Rankine engines, are rotary and orbital vane, as well as Lysholm-type expanders. Besides high temperature and lubrication difficulties, major problems are the high content of particulate matter in th exhaust of diesel engines and the wear associated with these particulates.

Experimentation in all of the above directions is in progress, but no manufacturer has thus far developed a workable solution to the small car turbocompounding problem. Based upon present knowledge and technology,

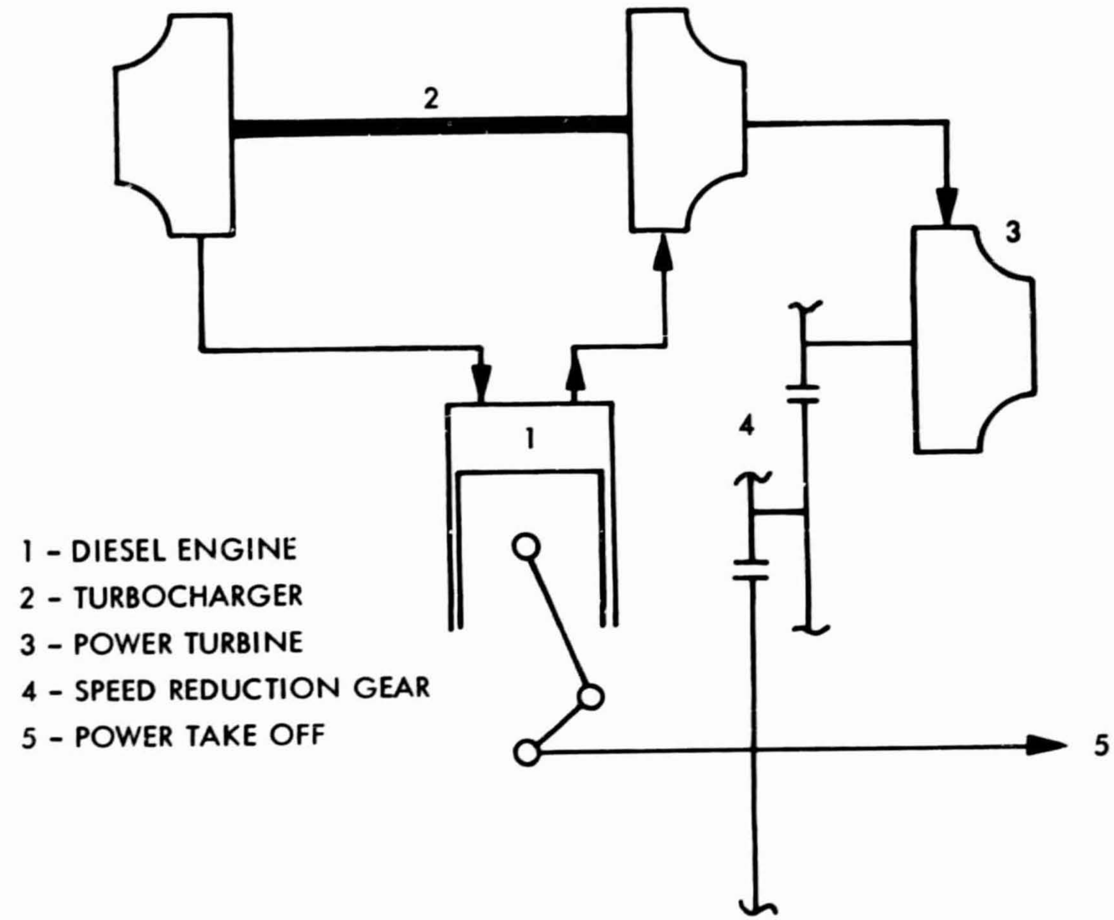


Figure 5.4-1. Schematic of Mechanically Turbocompounded Engine System (Example)

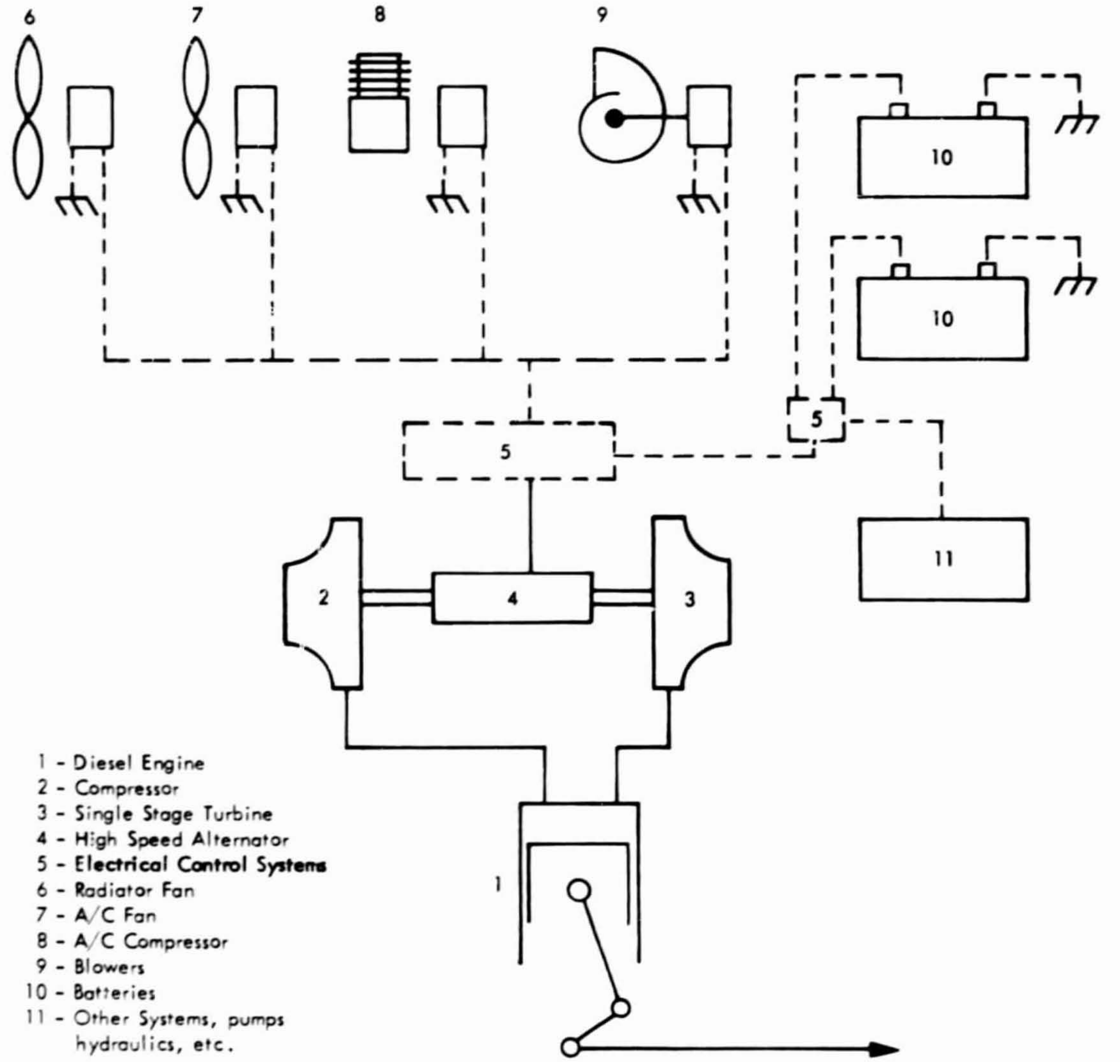


Figure 5.4-2. Schematic of Electrically Turbocompounded Engine System

electrical turbocompounding to operate engine accessories appears to be the most feasible approach for the near future. This will precipate down into the small engine field from large category diesel powered vehicles that have a relatively high demand for accessory power, such as air conditioned buses, special heavy duty and certain military vehicles. Reportedly, the producers of diesel-powered heavy duty vehicles are working in this direction.

5.5 RANKINE BOTTOM CYCLES

Because of a relatively high efficiency at moderate cycle peak temperatures (similar to those existent in diesel exhaust gases, 550 to 650°F), Organic Rankine cycles offer among the best potential for converting exhaust heat into usable power. DOE-sponsored work to develop Organic Rankine bottom cycles for use in heavy trucks is currently under way at Thermo Electron/Mack Truck, and at earlier times at Barber Nichols of Denver. The developers claim that the power of existing trucks can be increased by approximately 15% over a typical duty cycle without using additional fuel (Ref. 41).

However, as schematically shown in Figure 5.5-1, a relatively complicated system is required that does not lend itself easily to small car applications. A limiting factor in smaller engine sizes is the expander, which in the Thermo Electron system, consists of an impulse turbine geared into the main drivetrain. For smaller diesels, the use of a displacement type of an expander and a V-belt pulley drive appear to be a more feasible solution to the expander design problems although less efficient.

According to press information (Ref. 42), Chapman Engines, International, Inc., of Reseda, Calif. a small private developer, is experimenting in this direction with an Organic Rankine system for medium duty trucks that uses a chain driven, positive displacement, so-called "orbital", expander (Figure 5.5-2 and Figure 5.5-3). In contrast to conventional vane expanders, the rotor of the orbital expander does not rotate, but performs an orbital motion within the outer circular housing without changing its angular position relative to the housing. The vanes also do not rotate, but only sweep back and forth along the housing inner wall at a relatively low velocity. The Chapman expander is an attractive concept for application in small Organic Rankine systems, and also deserves consideration in turbocompounding approaches for small diesel engines in lieu of a small, high speed, power turbine.

Based upon present knowledge, the general concensus to date is that Organic Rankine systems are probably not commercially feasible in conjunction with diesel-engined passenger cars. The driving cycle of the passenger car also does not lend itself to exhaust heat utilization concepts because much time is spent at low load levels and idle speed.

5.6 COMBUSTION

The analytical and experimental tools required to study high speed combustion mechanisms, including those responsible for the formation of pollutants, have made fundamental combustion research a costly and time

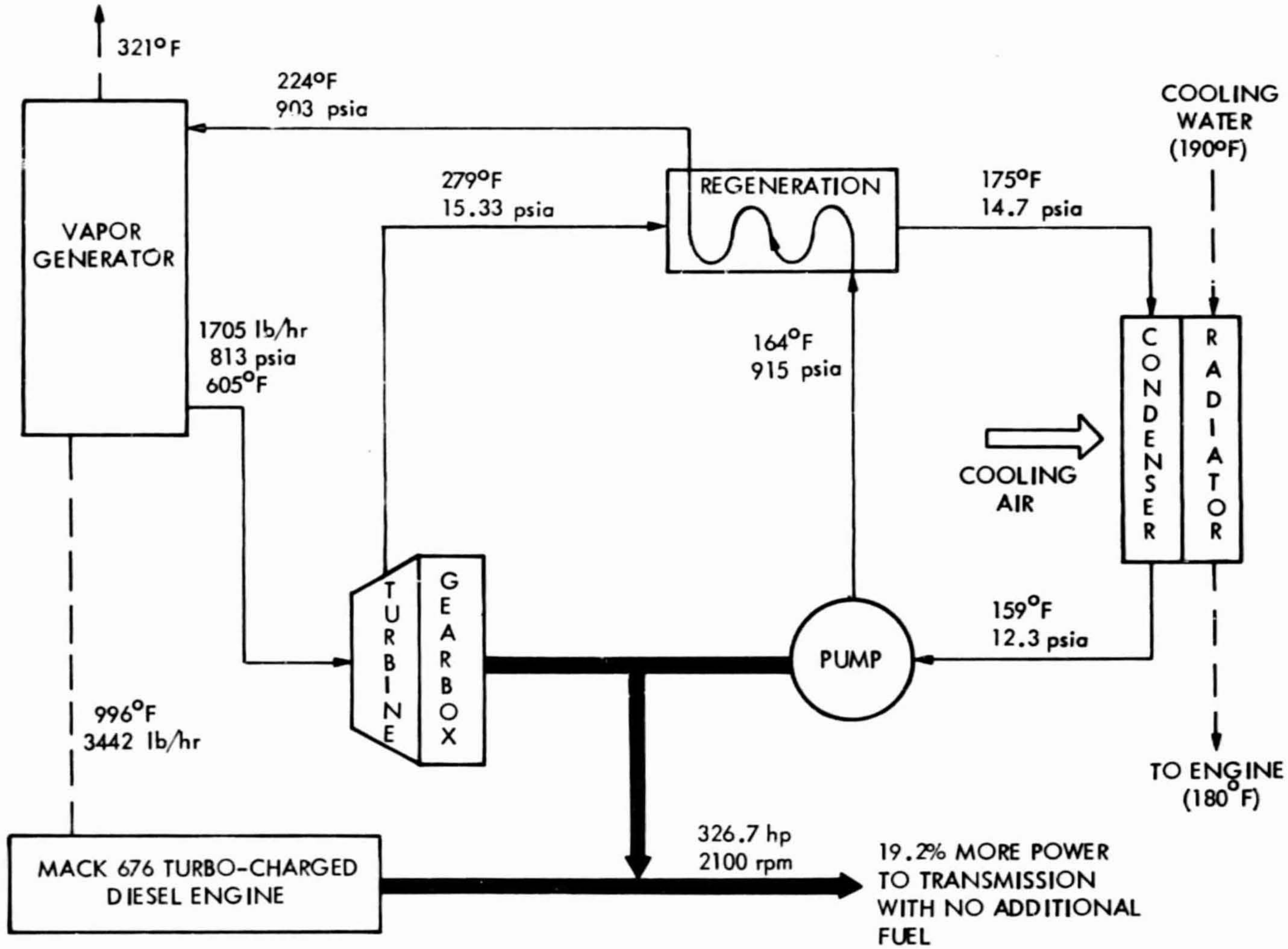


Figure 5.5-1. Diesel-Organic Rankine Compound Engine Schematic (Ref. 41)

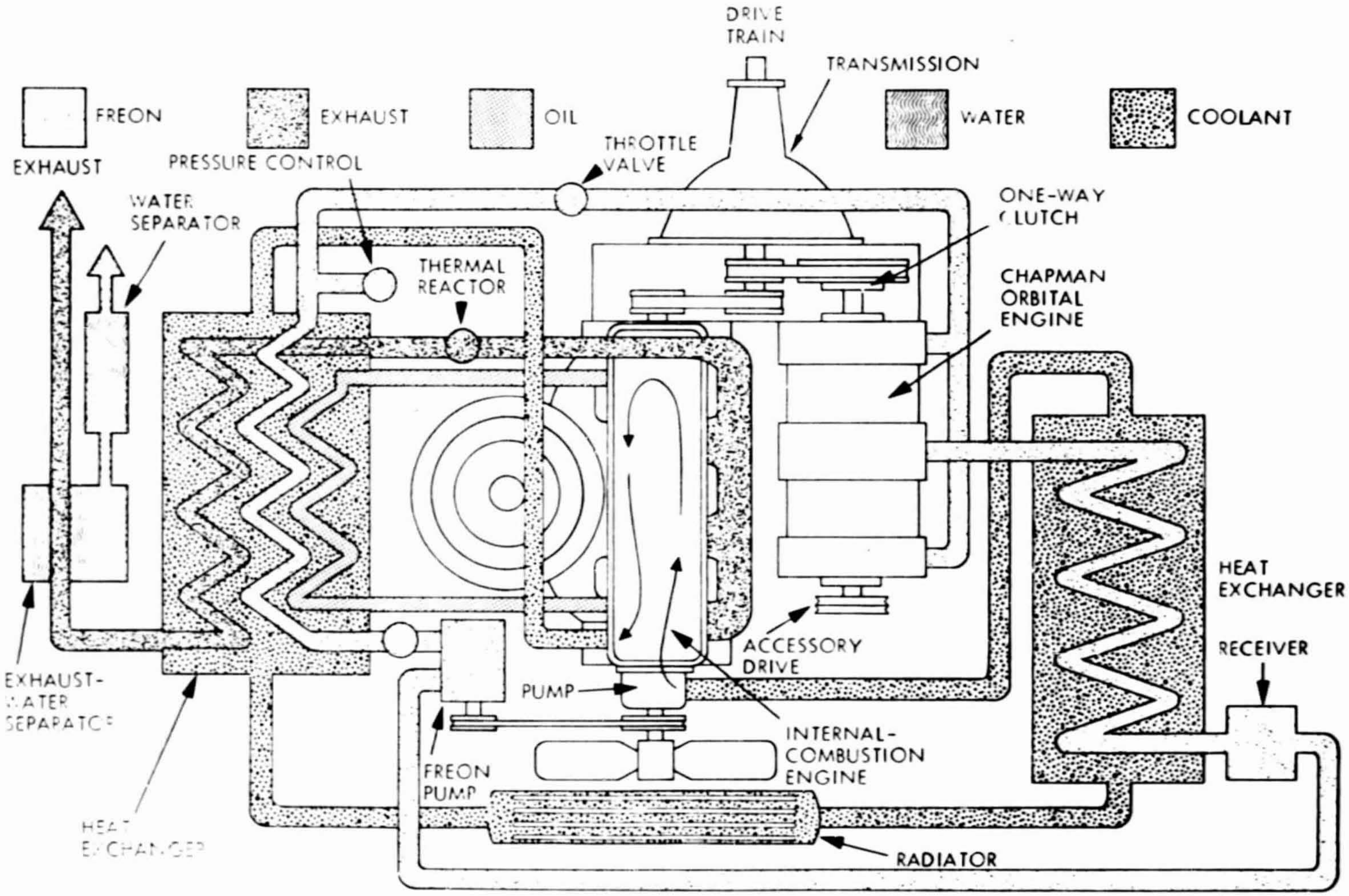


Figure 5.5-2. Cutaway View and Schematic of Chapman Engine (Ref. 42)

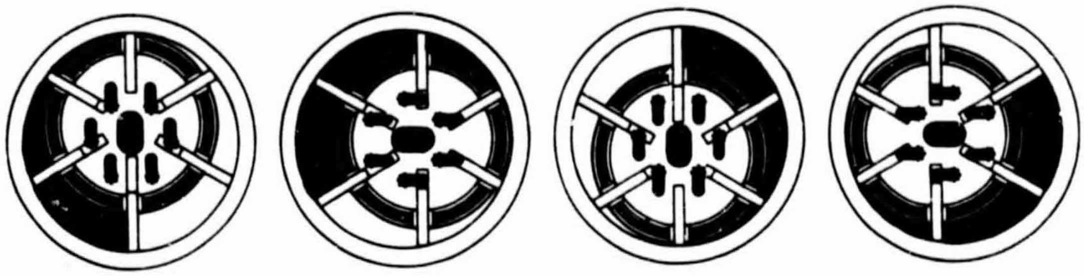


Figure 5.5-3. Schematic of Chapman Orbital Expander (Ref. 42)

consuming affair. Proposed research must consider the use of high speed equipment to visualize and to quantitatively evaluate combustion phenomena (Figure 5.6-1). Fuel injection flow and heat release rates, pressure and temperature changes as well as any other rapidly changing data that are necessary to interpret and to better understand the processes and interaction occurring in a high speed diesel combustion chamber must be accurately measured. A continuous recording of data at crankangle intervals of one degree or less is considered the minimum requirement for obtaining conclusive results, and for generating the data needed for realistic modeling of diesel combustion processes. Expensive special equipment is required for the characterization of gaseous and particulate emissions in regard to sizes, physical structure and composition.

Some items of primary interest for study are: (1) the effect of ultrasonic application on fuel atomization, mixing and flame propagation, (2) the use of high energy sources such as high intensity sparks, lasers, and plasma jet techniques to induce more homogeneous ignition, (3) the use of ceramic wall coatings capable of supporting a catalyst which will enhance the cracking and oxidation of unvaporized fuel, and (4) the introduction of electrical field effects to interpret and, possibly, to inhibit the formation and the size of particulates.

A definite need exists for fundamental research of this nature. It is expected that such efforts will produce striking solutions to diesel problems within the near future. Development now in progress is primarily concentrated on optimizing and refining the existing designs and injection systems, using conventional empirical and EPA accepted methods, to monitor the controlled pollutants NO_x, CO and HC, as well as smoke opacity and odor. The measuring and characterization of particulates are still unresolved problems, and HC formation and smoke opacity (Bosch Number) are used primarily, to judge particulate emission characteristics.

At the present time, the improvement potential of the divided chamber seems to be very limited, and a trend exists to return to the open chamber concept, which has attractive potential for improvement. As discussed earlier, the elimination of the pumping losses between chambers alone amounts to a gain of approximately 9% in terms of mean effective pressure and fuel consumption. This gain would compensate for losses caused by EGR, which must be introduced to reduce NO_x formation to the prescribed levels. The stratified combustion approaches currently taken to reduce NO_x in spark ignition engines are, in principle, also adaptable to the open chamber diesel. Approaches in this direction are under way.

Besides emissions, the primary problems with open chamber combustion are those of ignition delay and sudden initial heat release, which are mainly responsible for harsh operation and noise generation. Current approaches to the resolution of the open chamber performance problem go in the direction of using swirl and squish flow generated in the narrow clearance between the piston face and the cylinder head surface upon approaching the outer dead center. According to Figures 5.6-2 through 5.6-5, the apt use of squish flow has shown encouraging results in improvement of heat release, chamber pressure profile and NO_x with a relatively small reduction in fuel efficiency.

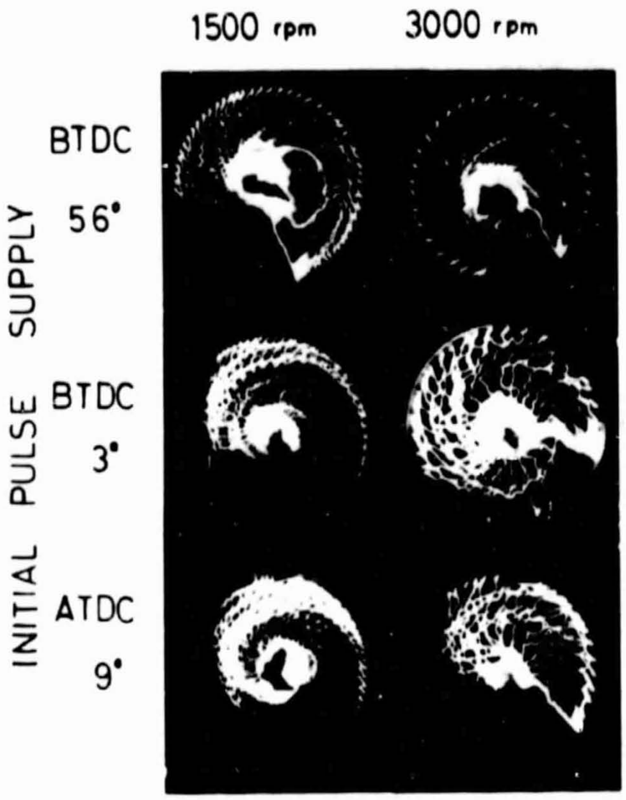


Figure 5.6-1. Visualization of Swirl Pattern in a Single Cylinder Diesel Combustion Chamber by Means of Sparktracing (Ref. 43)

Using similar techniques, the Kloeckner Humboldt Deutz (KHD) Company, Germany, has taken the approach to the two-stage combustion concept shown in Figure 5.6-5 (Ref. 45). Deutz claims that the concept is superior for reduced NO_x emission. Efficiency losses are on the order of 10 to 15% with EGR, as compared to a non-EGR direct injection engine. Deutz intends to introduce the concept into their line of air-cooled diesels marketed in the U.S. for light trucks as soon as emission standards dictate.

Fiat takes the approach to a three valve open chamber to obtain a high swirl and a good match between fuel injection and air motion over a wide speed range (Ref. 46). The fuel injector is located between the two smaller inlet valves, with one large exhaust valve opposite. Reportedly, CO and NO_x emissions obtained in tests with a Fiat 131 car match those obtained with swirl chambers, with hydrocarbons being relatively high (2 g/mi). The fuel economy range obtained was 42 to 52 mpg, not much better than that obtainable on cars with comparable weight and engines with current swirl chamber designs. Fiat claims that the direct injection diesel has a narrower operating range and requires more gears to use to its advantage.

As described in more detail in Section 3.6, BMW-Steyr has taken a daring step towards the solution of the open chamber, sudden heat release and diesel knock problem, by eliminating high pressure fuel lines and associated water hammer effects that greatly aggravate the accurate control of the fuel injection rate into the chamber. Instead of a central injector pump common to for all cylinders, the BMW-Steyr design provides for an injector integrated pump for each individual cylinder which, are assumed to be directly cam operated. This is not entirely new in diesel design history, but it never has been attempted for small, high speed multi-cylinder diesels. Because a number of other measures such as monoblock-design and engine insulation have been necessary for consumer acceptance, it can be assumed that the open chamber knock problem has only been partially resolved with the individual pump-injector design approach.

Renewed attention is also being given to an evaporative combustion concept demonstrated over 20 years ago by Maschinenfabrik Augsburg-Nuernberg (M.A.N.) in Germany. The design concept usually referred to as the "M" (Meurer) method, provides for a partially shielded valve to produce inlet swirl and a bowl-shaped combustion chamber which is deeply recessed into the piston (Figure 5.6-6A/B). The fuel injected into the chamber is not atomized but is squirted against the chamber wall and from there it disperses into a thin film that evaporates as combustion proceeds. Test engines modified to implement the concept have exhibited unusually flat torque characteristics (Figure 5.6-6C) and, as witnessed by the author of this report, the engine demonstrated smooth operation without the familiar diesel knocking over its entire speed range in an M.A.N. truck. A problem remaining is the thermal preconditioning of the bowl surface before continuous fuel evaporation actually takes place at a controlled rate.

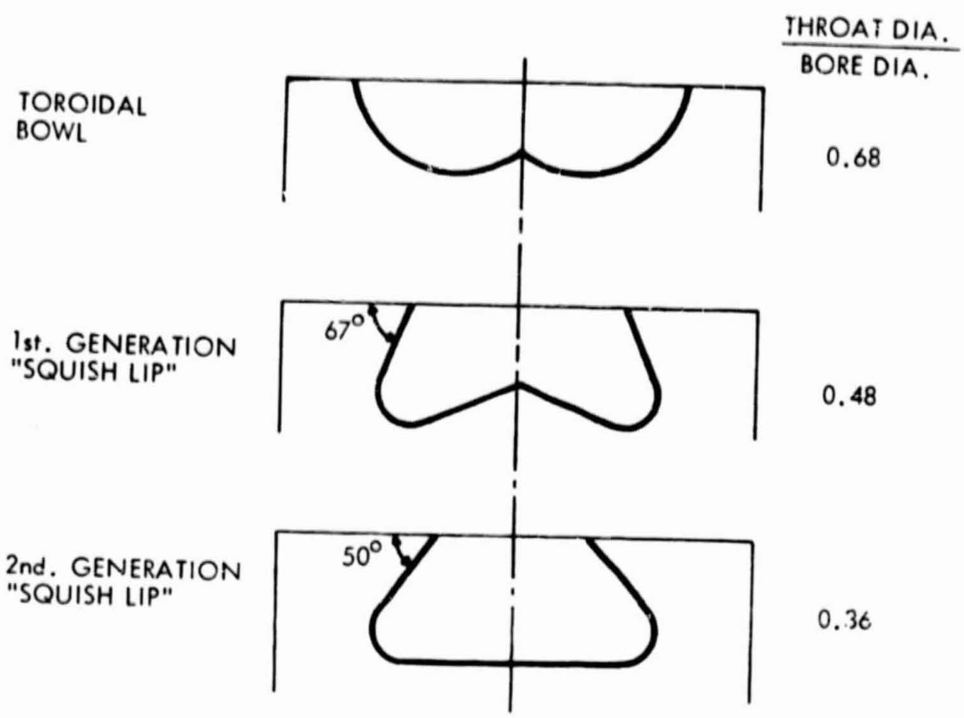


Figure 5.6-2A. Piston Bowl Profiles (Ref. 44)

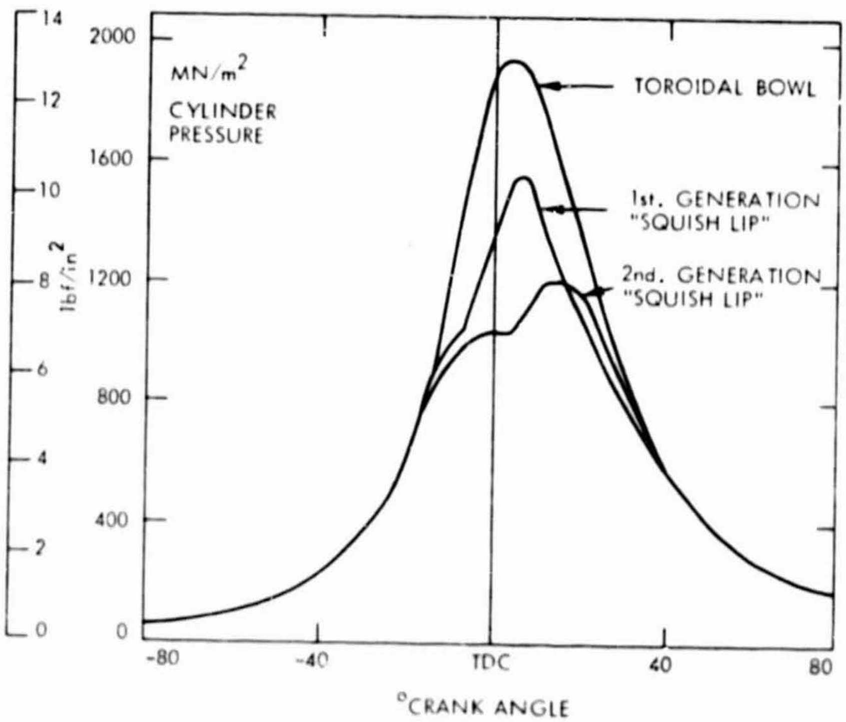


Figure 5.6.2B. Turbocharged Engine Cylinder Pressure (Ref. 44)

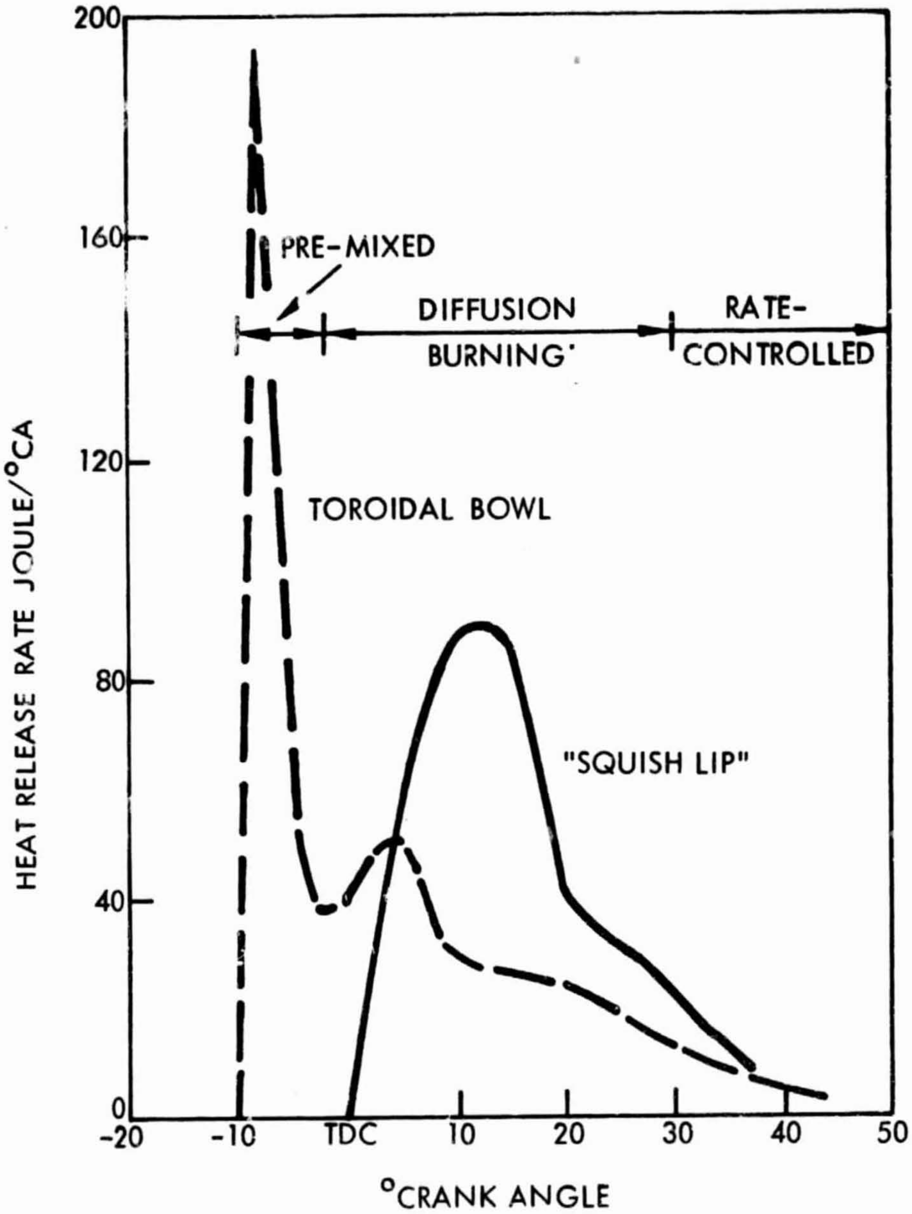


Figure 5.6-3. Heat Release Characteristics (Ref. 44)

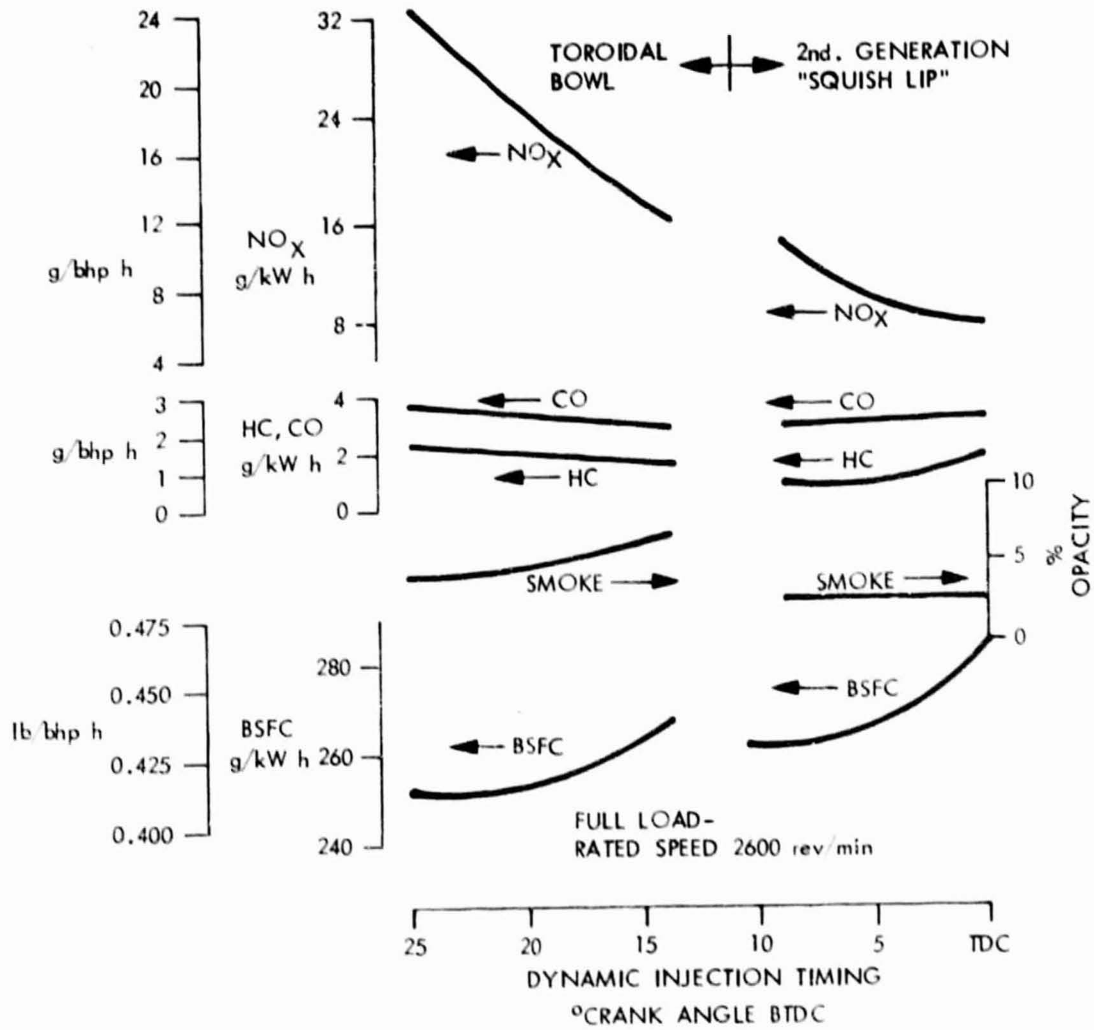


Figure 5.6-4. Effect of Timing on Performance and Emission Characteristics (Ref. 44)

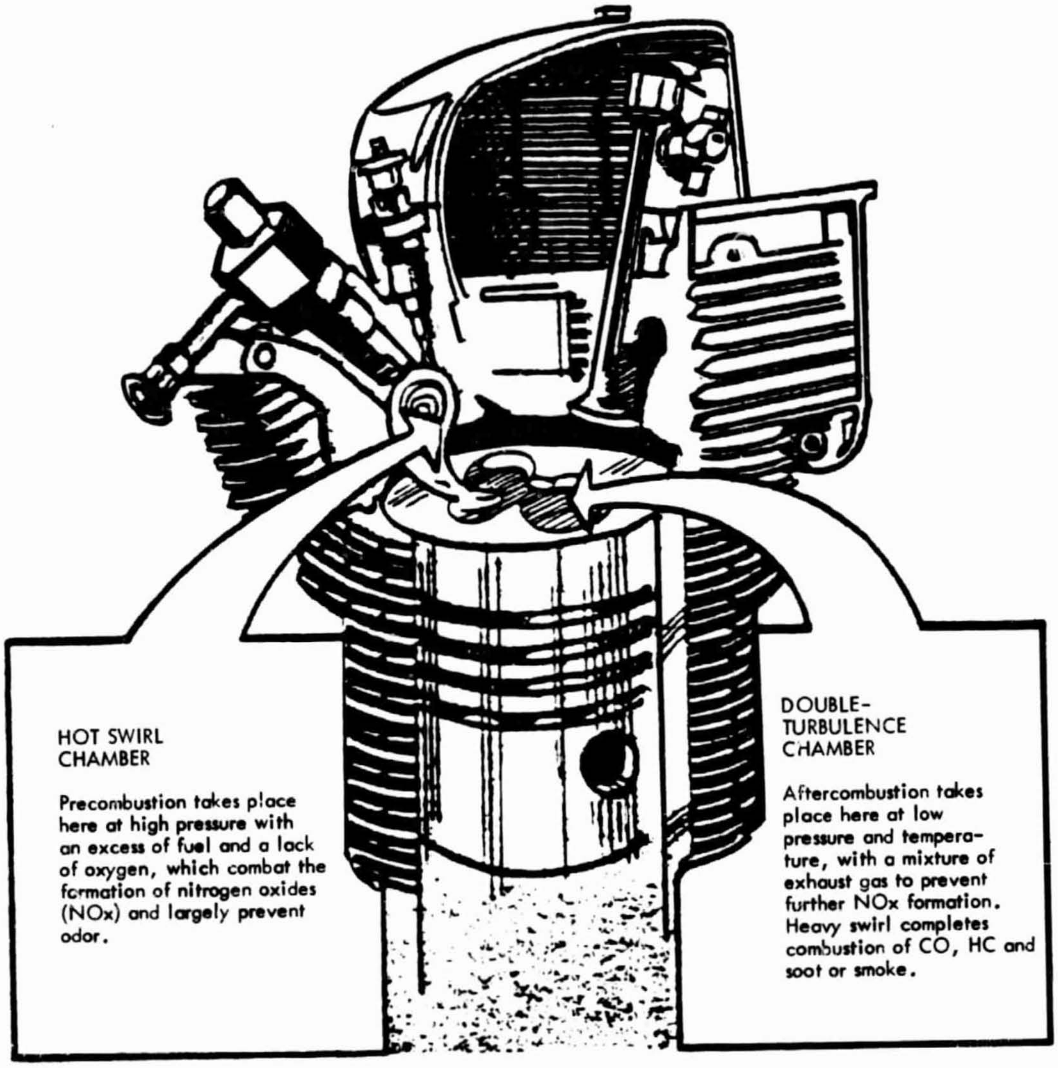


Figure 5.6-5. DEUTZ Two-Stage Combustion System (Ref. 45)

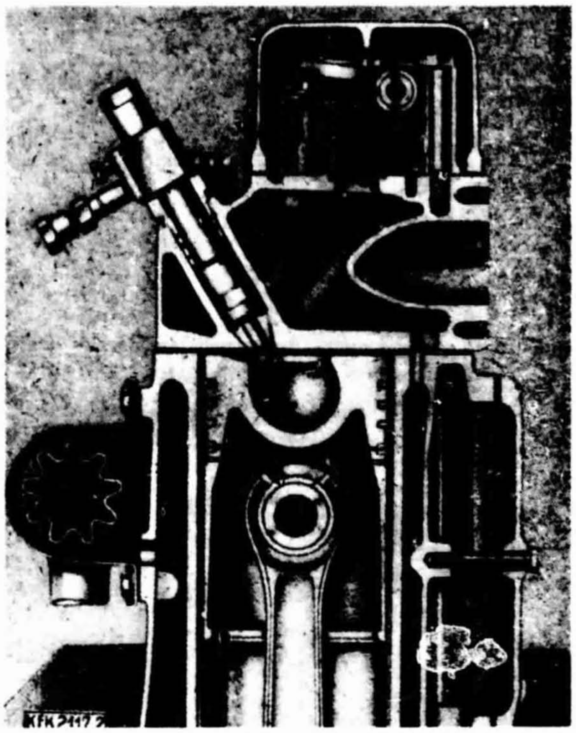


Figure 5.6-6A. M.A.N. Evaporative Combustion System - Cross-Sectional View of Test Engine (Ref. 47)

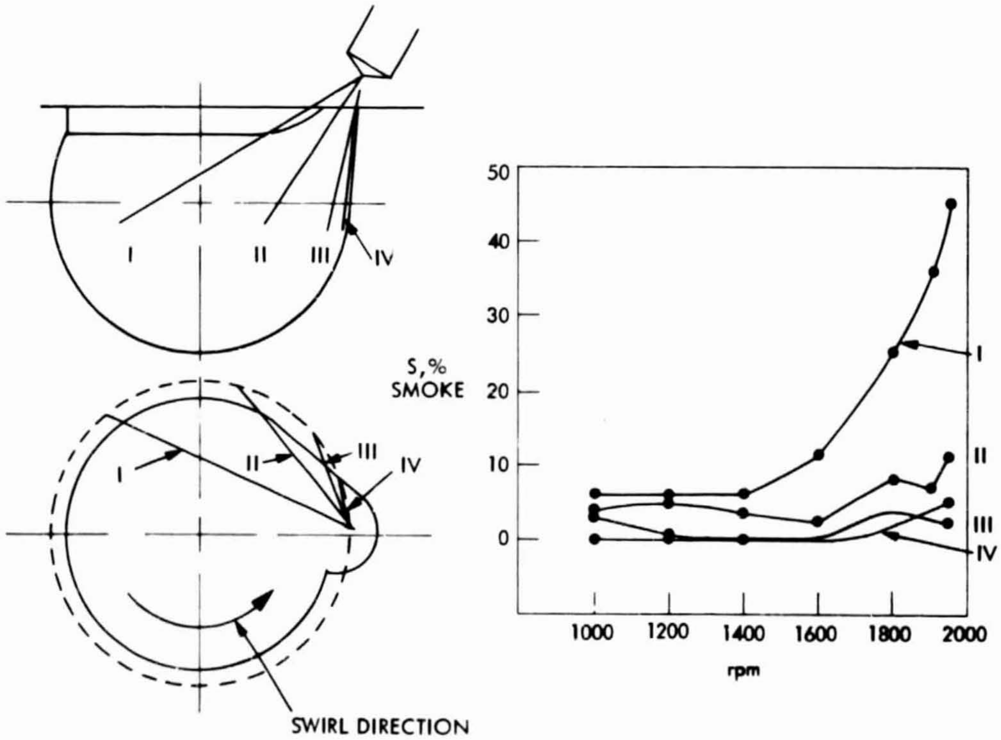


Figure 5.6-6B. M.A.N. Evaporative Combustion System - Effect of Fuel Jet Angle and Engine Speed on Exhaust Smoke (Ref.47)

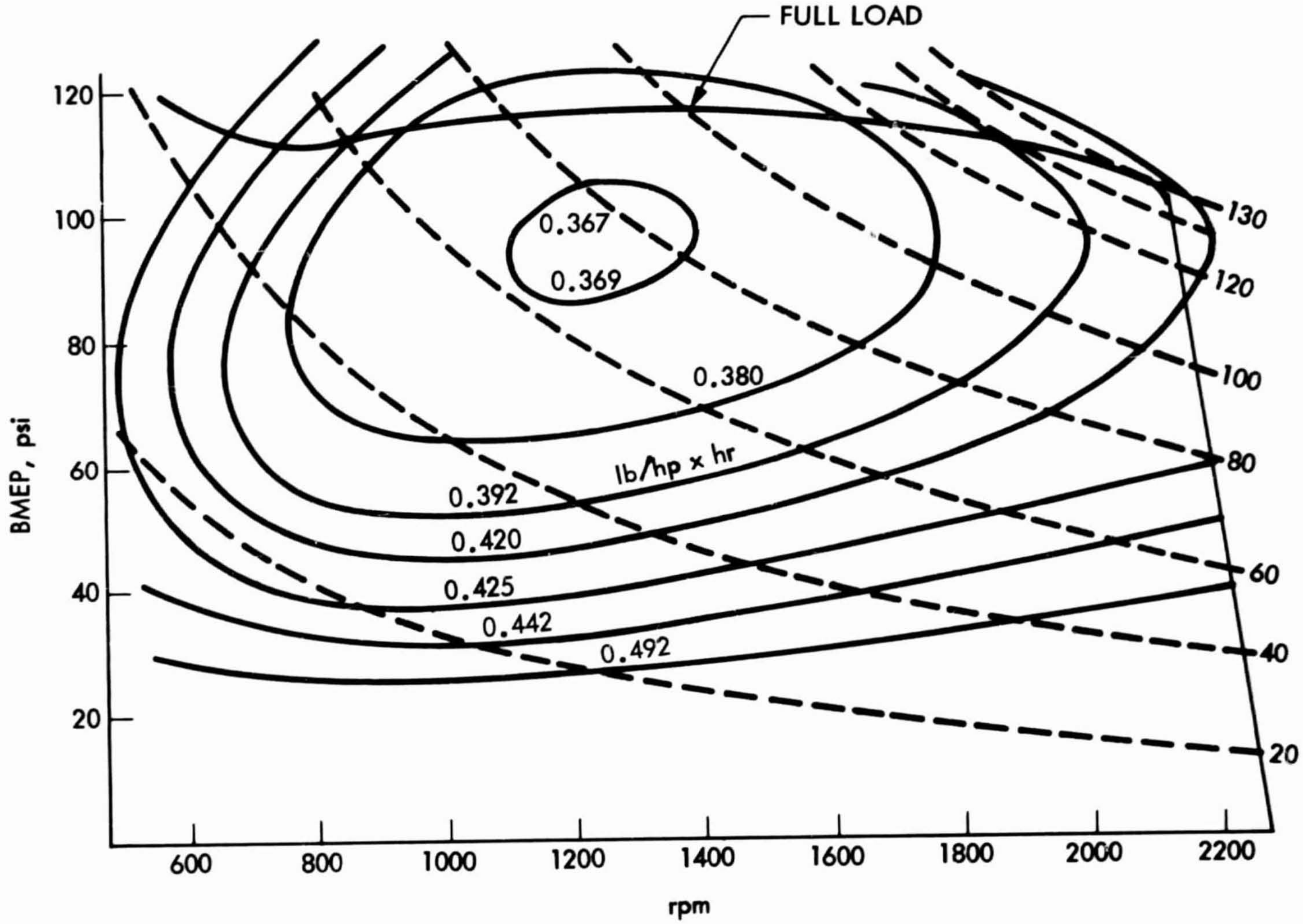


Figure 5.6-6C. M.A.N. Evaporative Combustion System Test Engine Characteristics (Ref. 47)

Also under renewed consideration are so-called semi-diesels (Figures 5.6-7, 5.6-8) which operate essentially as an open chamber diesel, with ignition brought about by a high temperature source before the self-ignition temperature is reached. The advantage is that lower compression ratios and broad cut fuels can be used. The Ford PROCO development is essentially aimed in this direction, although Ford avoids any reference to diesel-related features or developments, and refers to the system only as "PROgrammed COmbustion". As shown in Figure 5.6-9, fuel is metered to each cylinder in a diesel-like fashion by using individual injection nozzles at relatively low pressures (250 to 300 psig), starting as early as 70 to 90 degrees BTDC during the compression stroke. The compression ratio is low (approximately 11.5) requiring the aid of spark plugs to initiate combustion. Two-stage stratified combustion, an extremely wide air-to-fuel ratio and high EGR tolerance are brought about by using of piston bowl and squish flow techniques, high swirl producing inlet ports, and two spark plugs (Figure 5.6-10).

The system reportedly delivers a fuel economy almost comparable to that of a conventional pre-chamber diesel without the known disadvantages (noise, odor, etc.) (Ref. 48). However, the PROCO engine must use unleaded gasoline and an oxidation catalyst to meet emission requirements. This unfortunately eliminates some of the prime features that are attractive to the diesel-oriented consumer, such as lower maintenance cost, fuel storability (less flammable), and reliability.

The feasibility of the Ford PROCO concept has been demonstrated in extensive laboratory and vehicular tests. However, the implementation of the concept, will still require considerable development before becoming feasible with production hardware. It is not expected that PROCO will drive the diesel out of the market, but it might spur developments in the direction of semidiesels with compression ratios of about 15 to 16 that will accept a variety of broad cut fuels, rather than gasoline or diesel fuel alone.

5.7 VARIABLE COMPRESSION RATIO

The relatively high compression ratio of automotive diesel engines (21-22) is primarily dictated by cold start requirements. Once the engine has been warmed up, it would be more advantageous from the fuel efficiency and the NO_x standpoint to operate at a lower compression ratio of about 15. The cycle efficiency that can be gained from higher compression ratios is small and is usually cancelled by increased internal friction losses. Lower compression ratios are of particular advantage in turbocharged engines. For a given maximum system pressure, lower compression ratios permit application of higher inlet pressures, which increases the mass flow through the engine and allows for operation at generally leaner combustion with reduced formation of NO_x and smoke.

One approach to the implementation of a lower compression ratio followed by Cummins is a fixed geometry, low compression ratio diesel engine system that uses an inlet air heating system for engine start and for light load operation. Supercharging is accomplished by means of a Brown-Boveri Comprex

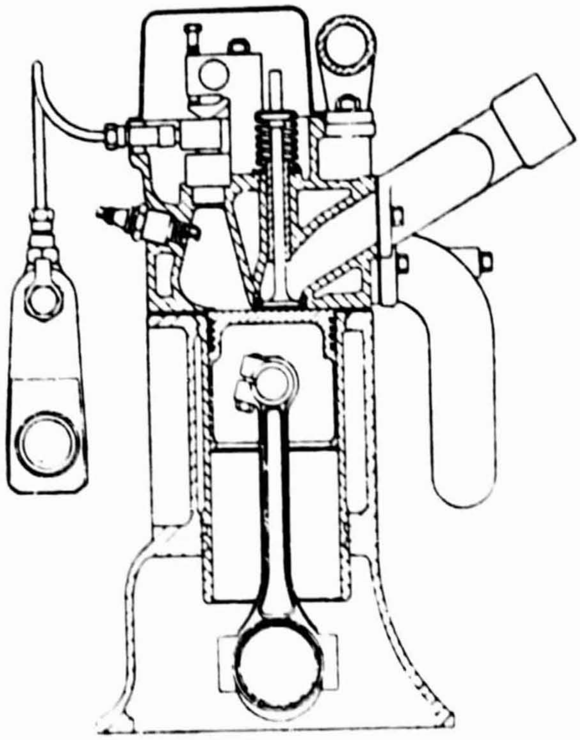
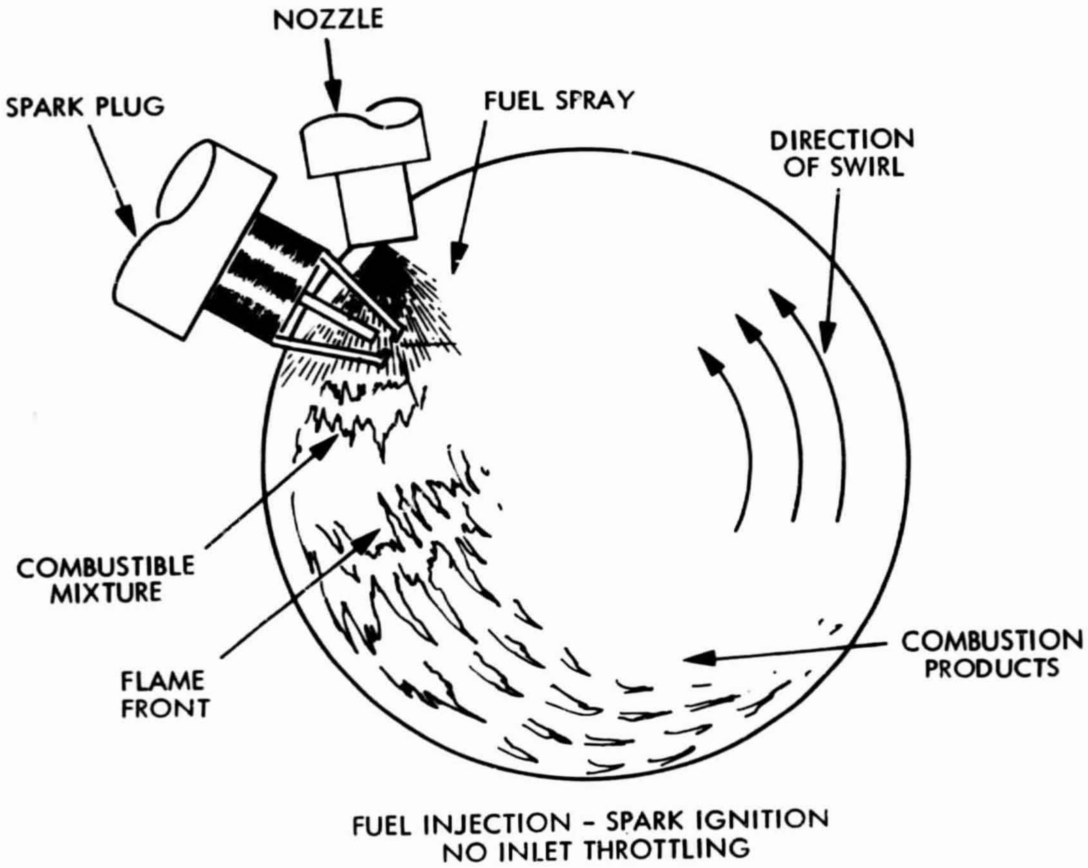
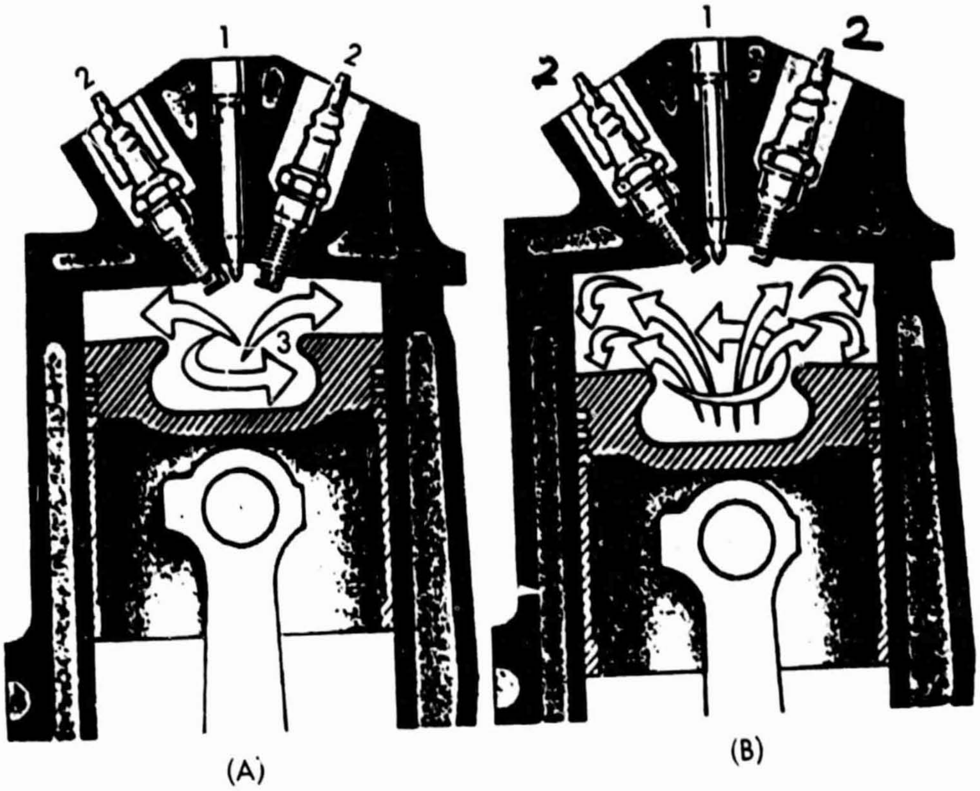


Figure 5.6-7. Hesselman-Based Semi-Diesel Engine by A.M. Starr (1948)



1. NO PREMIXING - FUEL INJECTION STARTS AT TIME OF IGNITION
2. SPARK IGNITES FIRST INCREMENT OF FUEL AS MIXTURE IS FORMED
3. FUEL IS BURNED AS RAPIDLY AS INJECTED

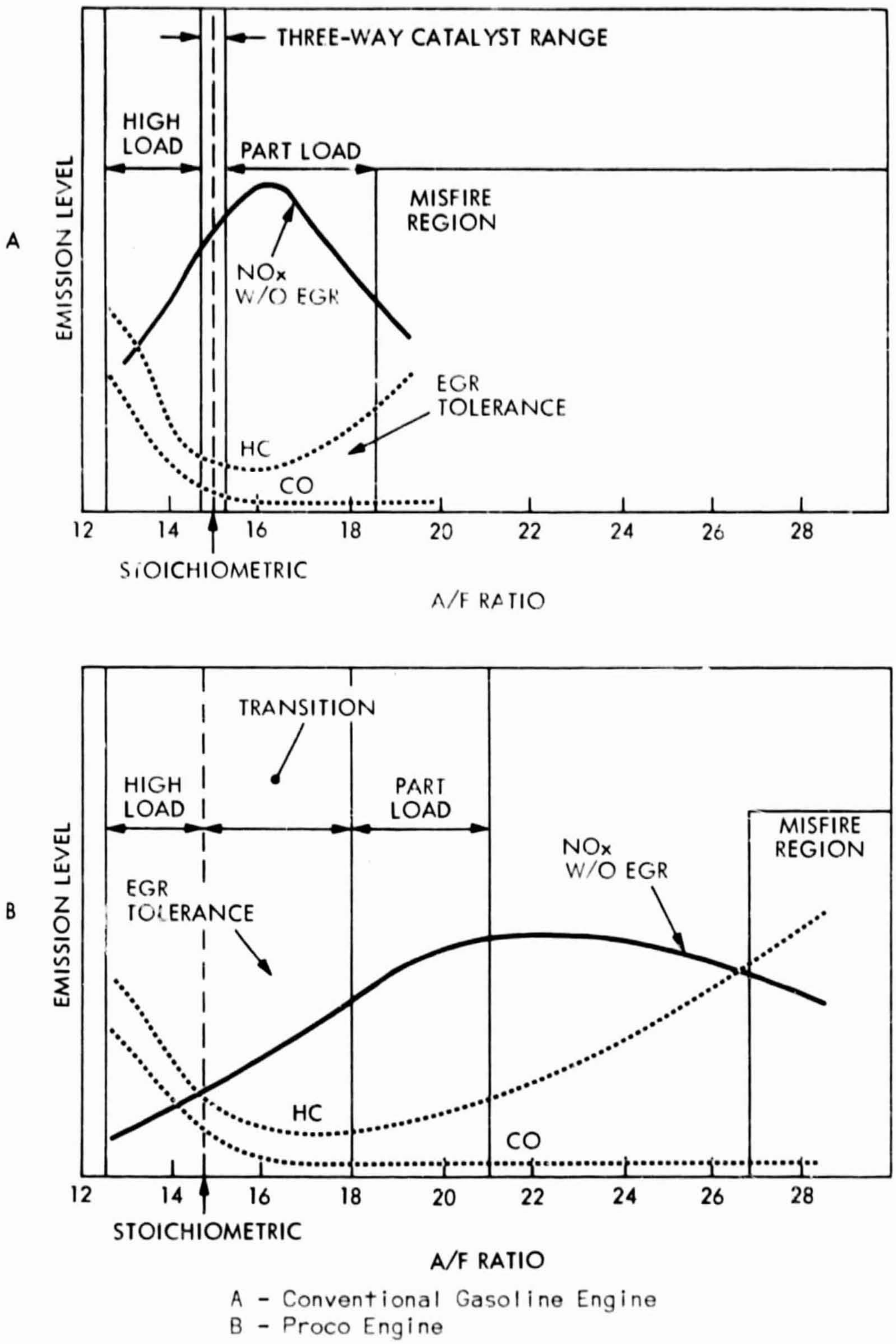
Figure 5.6-8. Schematic of Texaco Controlled Combustion System, Based on Semi-Diesel Engine (Ref. 45)



Typical Criteria:

Two-Stage Combustion
A - Rich Burn-First Stage
B - Lean Burn-Second Stage
1. Injection Nozzle
2. Spark Plugs - 2 each
3. Squish Lip Bowl

Figure 5.6-9. Schematic of Ford Proco Engine (Ref. 48)



A - Conventional Gasoline Engine
B - Proco Engine

Figure 5.6-10. Comparison of Combustion Characteristics (Ref. 48)

type pressure exchanger. Using the unavoidable entrapment of exhaust gases to its advantage, the comprex cell-wheel-charger also doubles as an EGR system. Cummins claims that the system is expected to have excellent low speed fuel economy, transient response, and NO_x and smoke characteristics. They refer to the system as an Otto-competitive diesel engine particularly suited for light vehicle applications (see Figure 5.7-1, Ref. 39).

Other approaches use a variable compression ratio capability to cover the variety of operating conditions between cold start and economical operation. A design patented by the British Internal Combustion Engine Research Institute under development at Teledyne Continental Motors accomplishes a change of compression ratio during engine operation by using a self-regulating, hydraulically actuated piston assembly which is schematically shown in Figure 5.7-2. The position of the outer structure "A" is controlled by the amount of oil in the upper and lower chambers. The movement between the outer structure "A" and the inner structure "B" brought about by the force exerted on the piston crown by the chamber pressure is controlled by a fixed orifice and by a relief valve, from where the oil is discharged back into the engine crankcase from an opening in the lower end of the piston. If the peak combustion pressure exceeds a pre-set value, the piston crown moves closer to the pin and reduces the compression ratio according to the method described. The designed-in minimum compression ratio is on the order of 12.

This approach is well conceived in principle and has been proven feasible by test. However, taking the thermal problems of pistons into account, this approach does not appear too feasible in a supercharged engine, where a variable compression ratio would be most desirable. It is doubtful that hydraulic continuity can be maintained in such a system under thermally demanding operating conditions unless a sufficient amount of oil is continuously flushed through the piston to keep engine and oil temperatures below a critical level to prevent oil evaporation or coking.

Figure 5.7-3 shows a different approach which is not very attractive from the weight and size standpoint, but is readily applicable to two-stroke opposed piston engines without interfering with critical components such as the pistons (Ref. 50). The concept of using rocker arms instead of two separate crankshafts is a conventional though not very popular design, but lends itself to the implementation of variable compression ratio by simply shifting the pivot points of the rocker arms. The design shown in Figure 5.7-3 accomplishes this by using eccentrics that allow variation of the compression ratio under load from 6 to 21. In the case shown, changes in the compression ratio are also associated with desirable changes in port areas, wherein the effective port area and the mass flow increase as the compression ratio decreases.

The concept has been tested at the University College in Dublin in a single and in a three-cylinder version. Encouraging results have been obtained, some of which are shown in Figures 5.7-4, A/B/C.

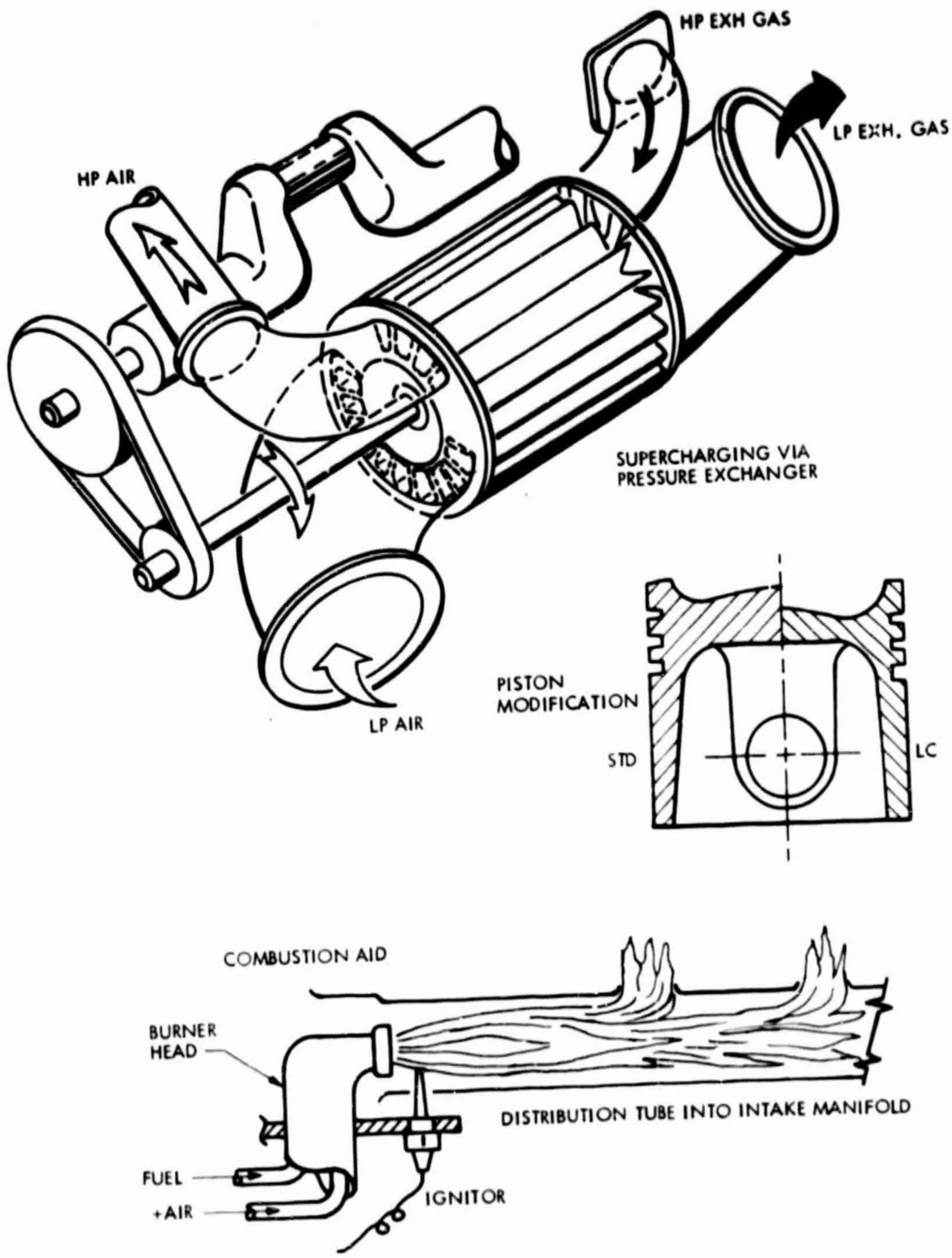


Figure 5.7-1. Basic Low Compression Ratio (LCR) Engine Concept (Ref. 39)

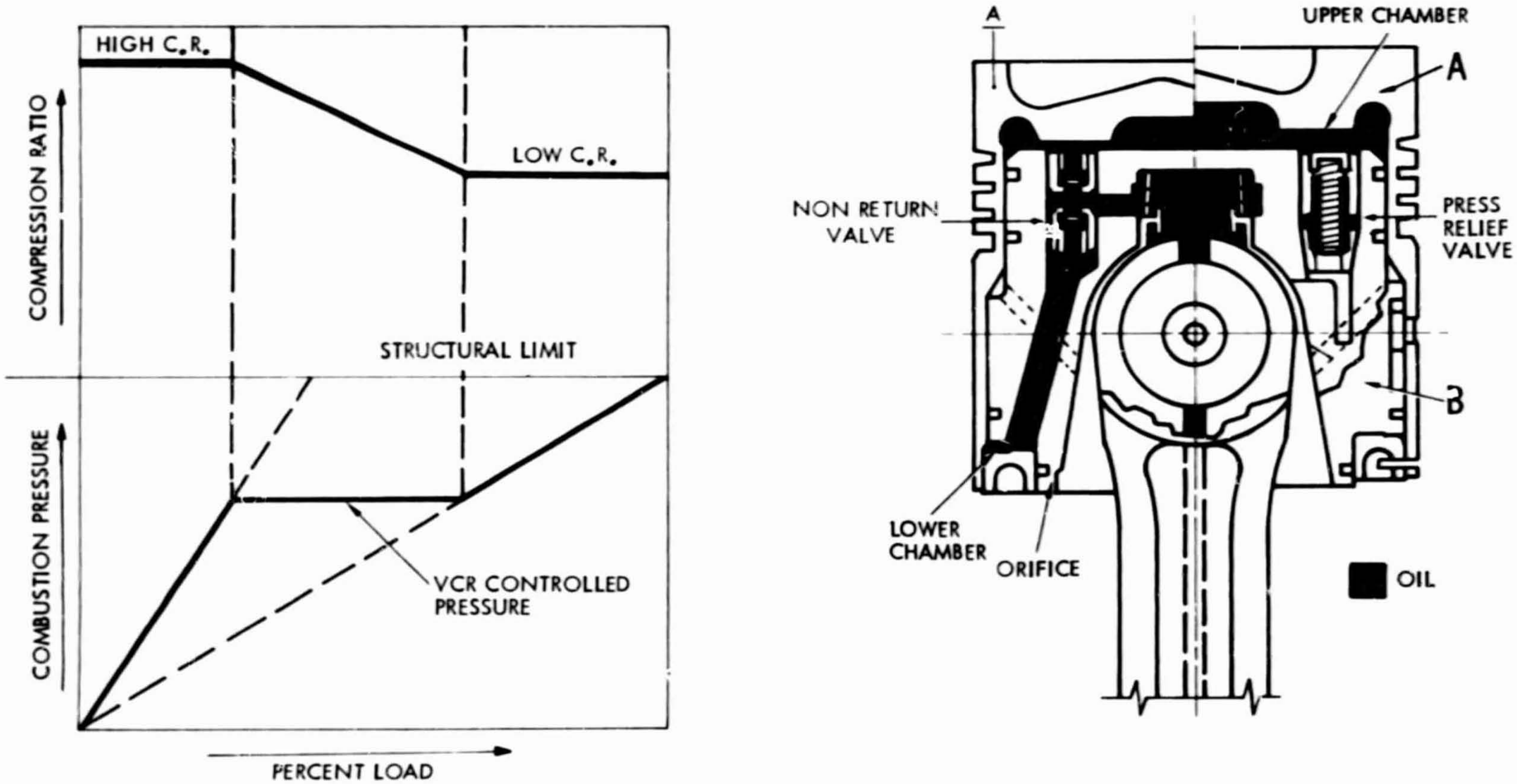


Figure 5.7-2. Variable Compression Ratio Piston
(Ref. 49)

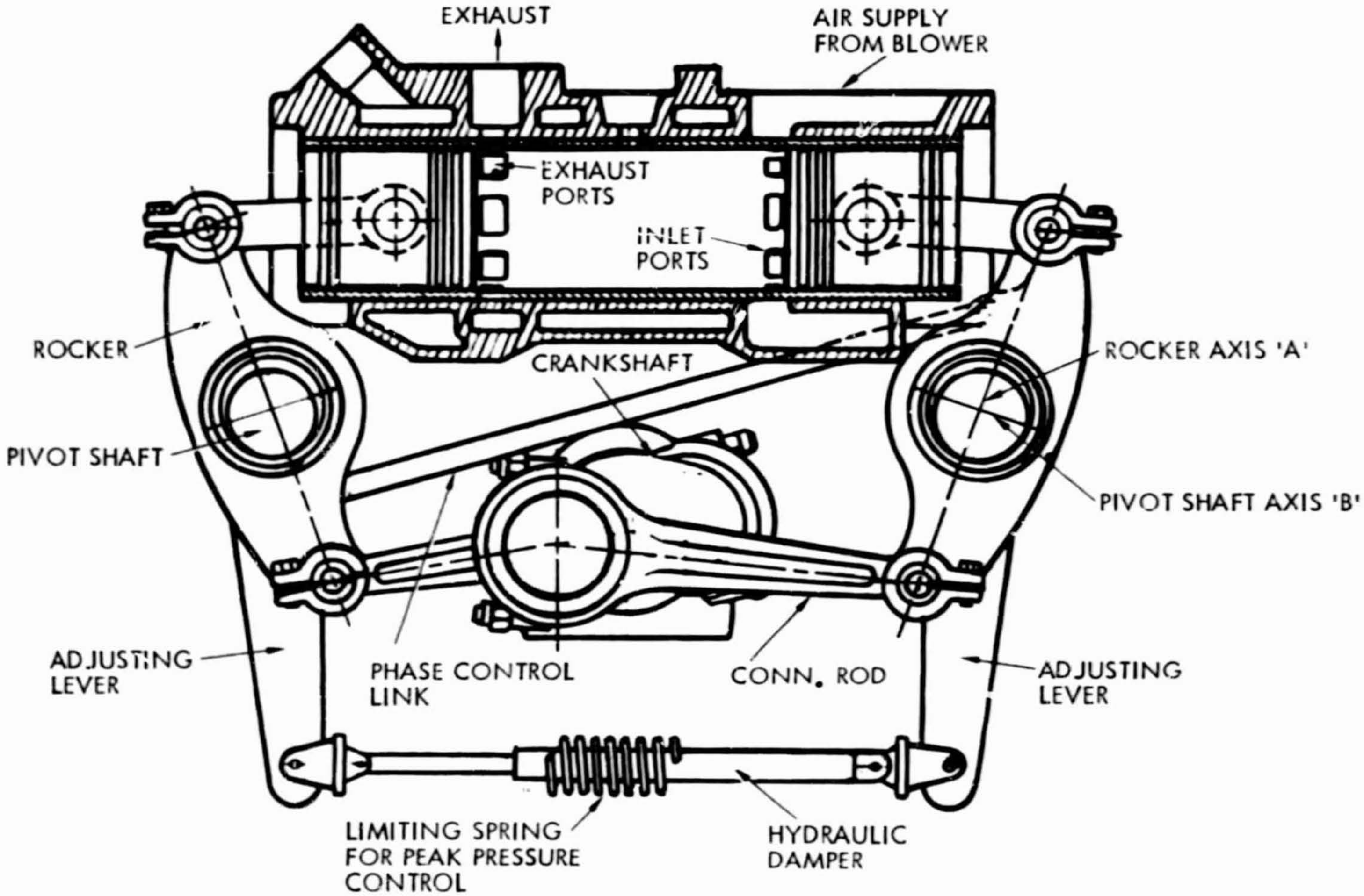


Figure 5.7-3. V.C.R. Opposed Piston Engine (Ref. 50)

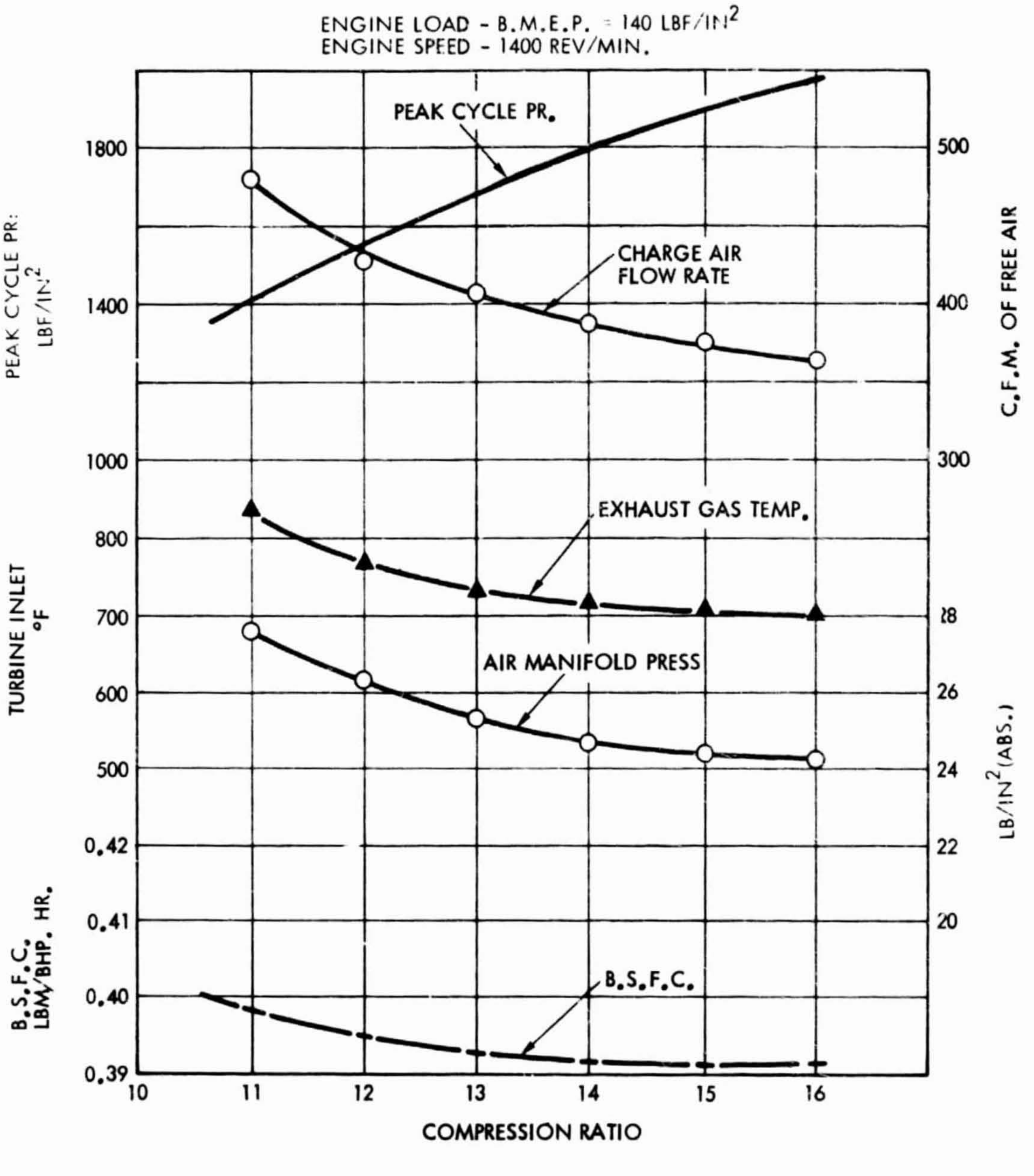


Figure 5.7-4A. Effect of V.C.R. at Low Load (Ref. 50)

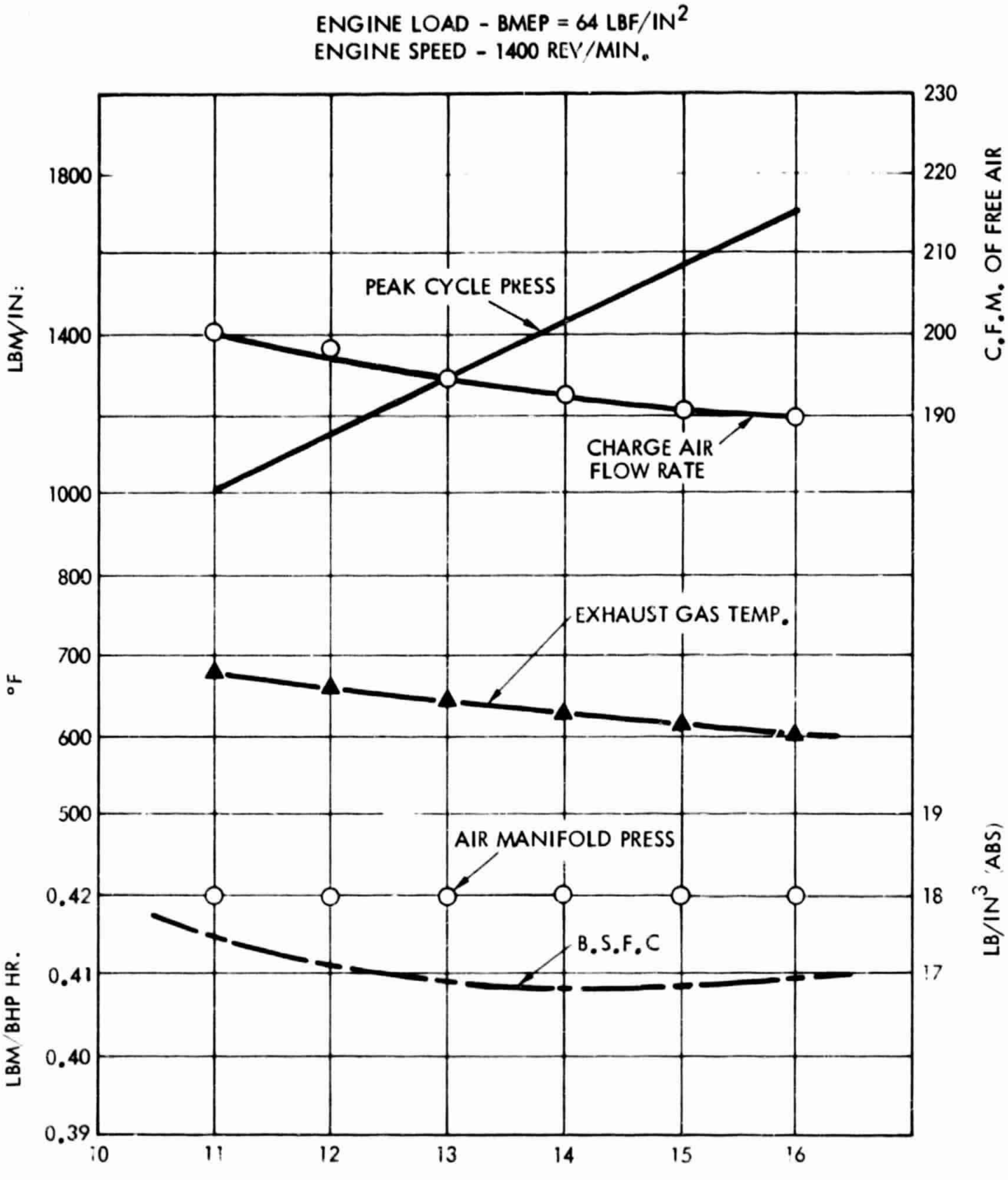


Figure 5.7-4B. Effect of V.C.R. at High Load (Ref. 50)

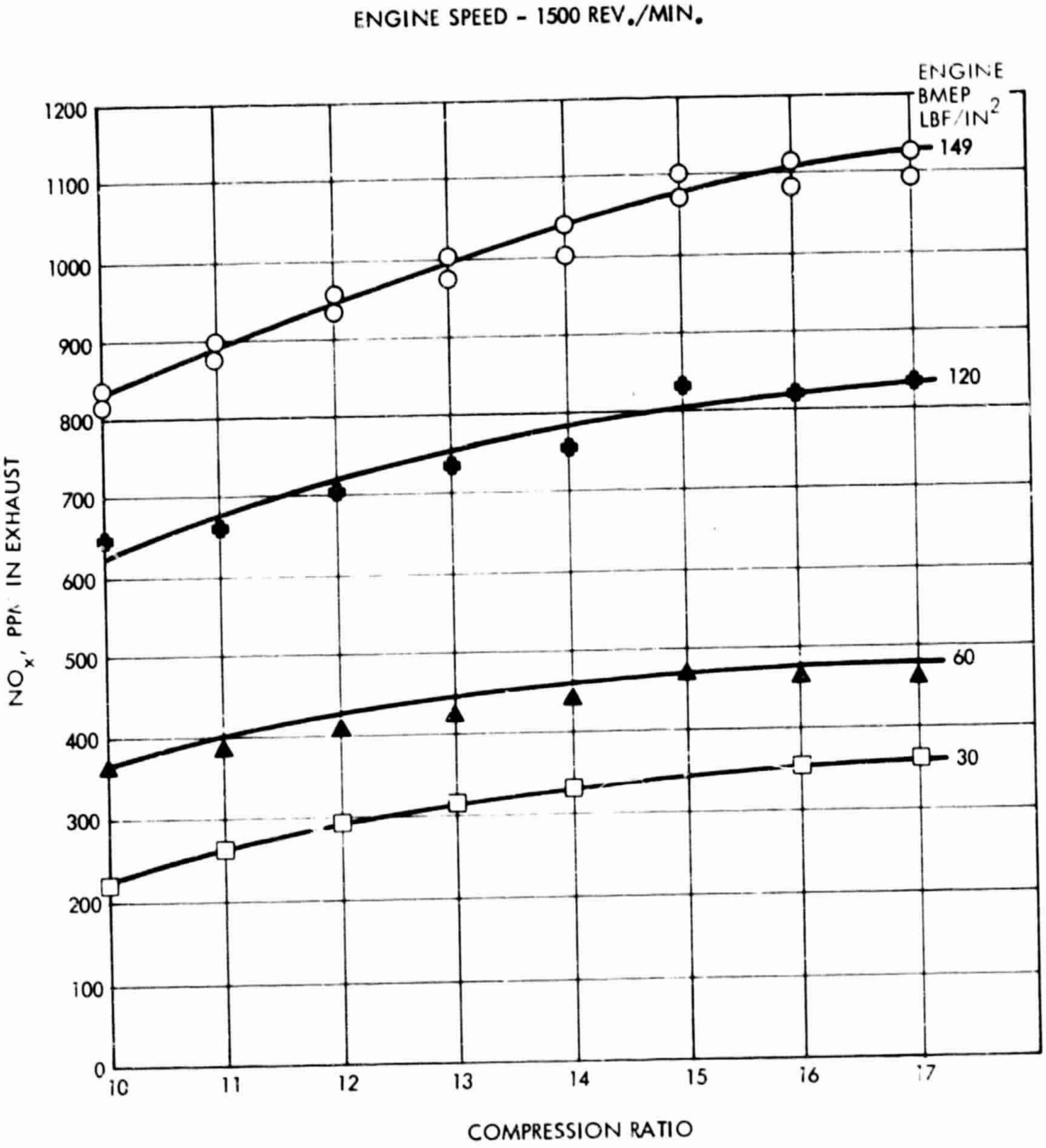


Figure 5.7-4C. Effect of V.C.R. on NO_x Formation (Ref. 50)

5.8 HEAT LOSSES

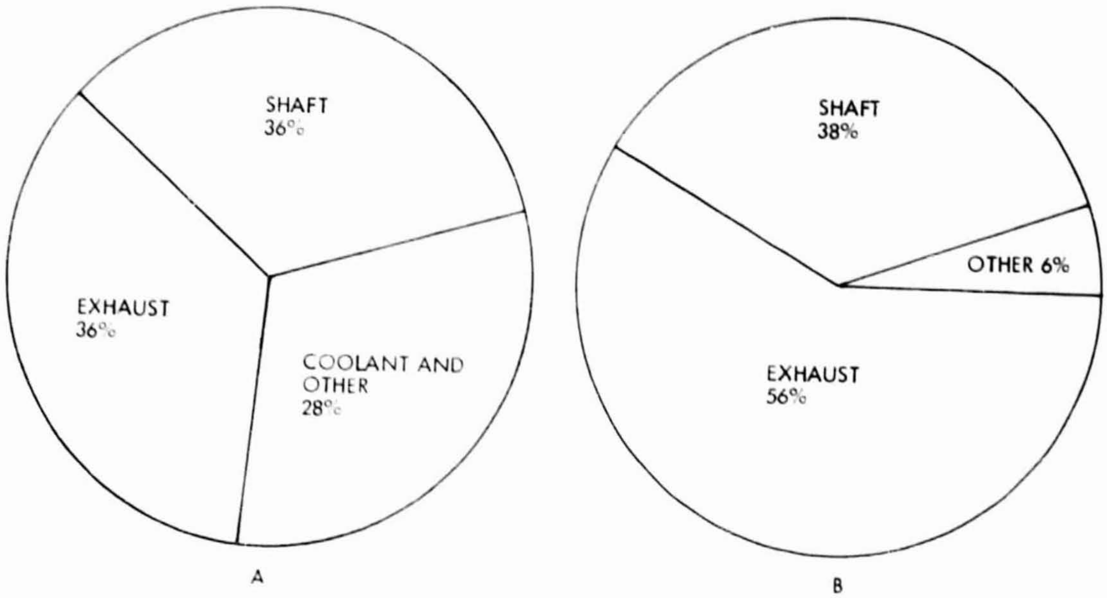
Reduction of heat losses presents the largest potential for improvement in terms of energy savings (Figure 5.8-1). However, advanced high temperature materials and new design technology are required. Current heat loss reduction efforts are aimed at gradually introducing insulated engine components of high temperature ceramics such as silicon nitride and silicon carbide. This would progress essentially in order of feasibility, starting with stationary parts such as the pre-chamber and cylinder head, then going to the piston crown, piston, and perhaps the cylinders. The latter would require that the hot lubrication problem also be resolved. Conceivable methods are the use of graphite dust or gas bearings, for example. The so-called "adiabatic engine", schematically shown in Figure 5.8-2, combined with the elimination of the cooling system is the ultimate goal.

Insulating the combustion chamber will help eliminate incomplete combustion problems associated with wall quenching and also shorten ignition delay times. It is also expected to have a number of other beneficial effects on diesel operation such as tolerance of a wider range of fuels, lower compression ratios, a reduction of white smoke, combustion noise, odor, particulates, HC and CO. An efficient extraction of mechanical energy from the exhaust heat (turbocompounding or other means of waste heat use) is required to realize any significant energy savings. Otherwise, engine insulation will primarily raise the exhaust temperature.

Figure 5.8-3 shows the NO_x/fuel efficiency relationships for a mechanically turbocompounded adiabatic engine as compared to an intercooled, supercharged, direct injection diesel engine of current design, as estimated by Cummins (Ref. 39). Combustion parameters, such as temperature, dwell time and excess oxygen, are mainly responsible for the formation of NO_x, but because they are only slightly influenced by heat wall losses, the minimum NO_x level that can be obtained with both engines is the same. However, large reductions in fuel consumption of up to 25% can be obtained if comparison is made at a constant NO_x emission level.

The strength of high temperature ceramic materials have been greatly improved during recent years, but they are still relatively brittle, which makes use in reciprocating engines somewhat more difficult than in gas turbines. In a turbocharged diesel, the structural members which establish the boundaries of the combustion chamber must withstand pressures on the order of 2000 psi and must undergo up to one billion heat cycles over the life span of an engine. The use of ceramics in engines is still in its infancy, but encouraging results have already been obtained. Cummins, a leader in the field of diesel ceramization, has taken firm action by directly addressing the problems associated with the most critical component - the piston. Others, such as Eaton, are concentrating on the development of other ceramic engine components, valves in particular.

Table 5.8-1 compares the properties of silicon nitride to those of conventional piston materials. Figure 5.8-4 and Table 5.8-2 show the piston design approach taken by Cummins, with related material temperature properties and stress information. The design consists of a ceramic top bolted to



A - Water Cooled Engine
B - Insulated (Adiabatic) Engine

Figure 5.8-1. Diesel Engine Heat Balance (Ref. 39)

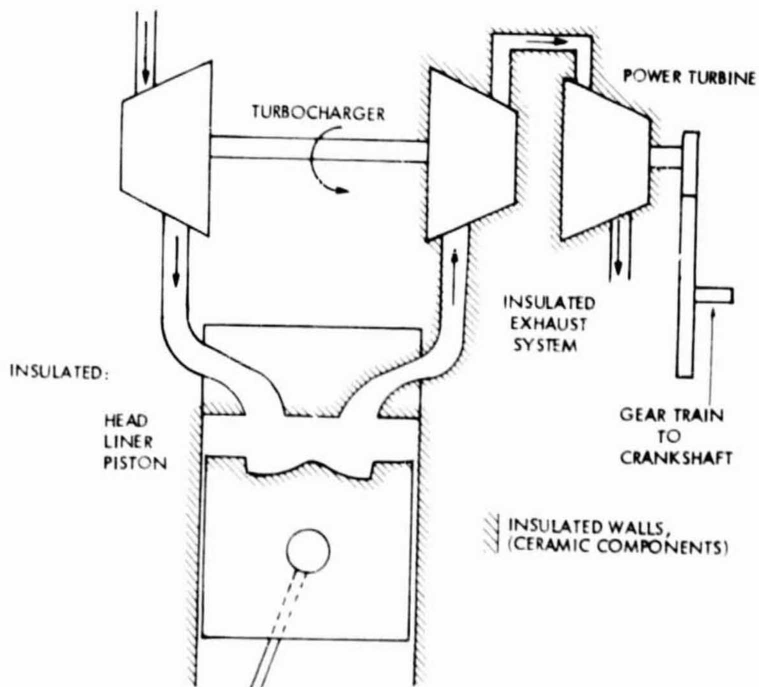


Figure 5.8-2. Schematic of Adiabatic Diesel Engine (Ref. 51)

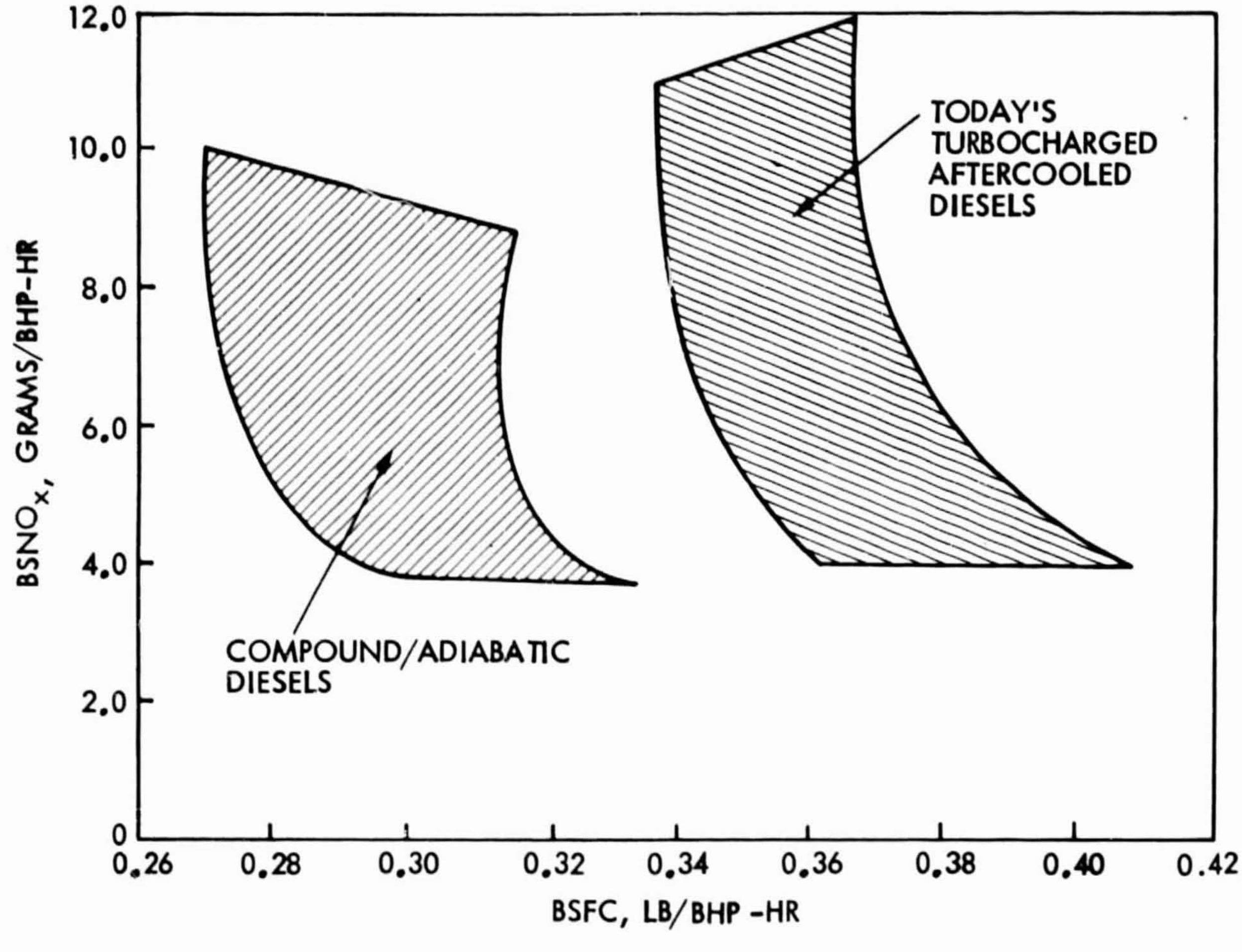


Figure 5.8-3. Comparison of NO_x/Fuel Economy Relationships (Ref. 39)

Table 5.8-1. Comparison of Properties of Silicon Nitride Ceramics and Conventional Piston Materials (Ref. 52)

	Aluminum Alloy		Grey Cast Iron	Modular Cast Iron	Silicon Nitride	
	LM13[a]	LM26[b] (SAE 332)	BS 1452	BS 2789	HPSN[c]	RBSN[d]
Density, Mg/m^3	2.70	2.74	7.3	7.2	3.2	2.2-2.75
Melting Point, °C	~570	~550	950-1230	950-1230	1750-1900 decomposition	
Thermal Expansivity $\times 10^{-6}$/°C, Linear, 20-1000°C	19	20	12	11	2.5	
Thermal Conductivity, W/mK (at room temperature)	117	104	50	29	25-35	8-40
Specific Heat Capacity, J/kg K	900	900	450	450	700	700
Strength, M Pascal	190-280	250	200-300	370-725	800[e]	230[e]
Elongation (%)	~0.5	1.3	-	17.2	-	-
Young's Modulus, G Pascal	71	80	103	166	300	175
Hardness	150-80HB[f]	100HB	180-260HB	250HB	1675-1950[g]	900-1350[g]

Analysis %:	Cu	Si	Mg	Ni	Fe	Mn	Ti	Zn	Sn	Pb
[a] LM13	0.7-1.5	10-12	0.8-1.5	0.7-1.5	1.0	0.5	0.2	0.5	0.1	0.1
[b] LM26	2.0-2.4	8.5-10.5	0.5-1.5	1.0	1.2	0.5	0.2	1.0	0.1	0.2

[c] HPSN Hot Pressed Silicon Nitride (fully dense)
[d] RBSN Reaction-Bonded Silicon Nitride (incompletely densified)
[e] 3-Point Bend Strength
[f] HB Brinell Hardness
[g] Knoop Hardness, 100 g load, kg/mm^2

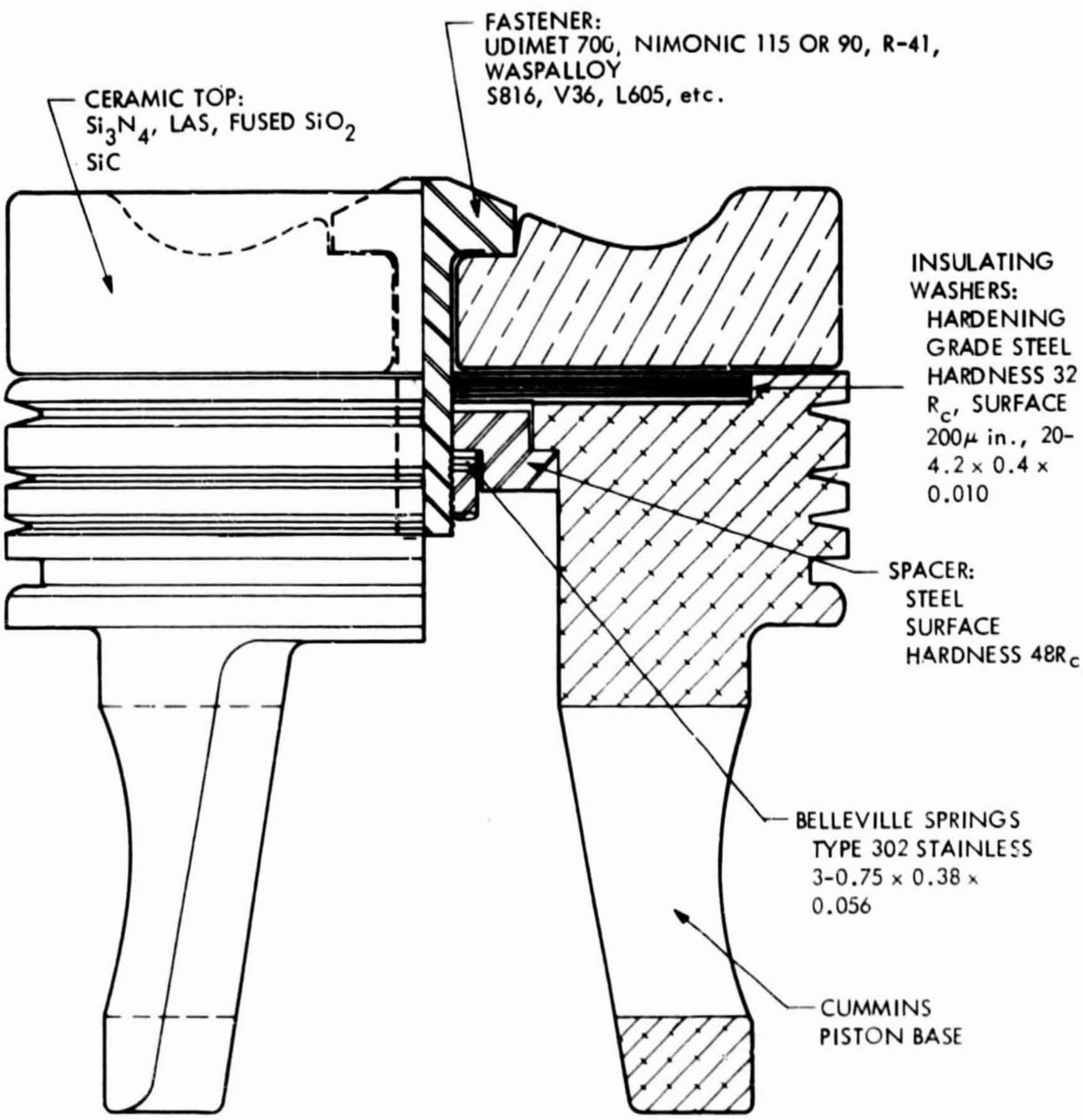


Figure 5.8-4. Adiabatic Engine Piston Concept Under Study at Cummins Diesel (Ref. 51)

Table 5.8-2. Stresses in Cap of Ceramic Piston as Estimated by [illegible] ef. 51)

Material Type	Coefficient Thermal Expansion $\times 10^{-6}/°F$ (1000°F)	Thermal Conductivity Btu/hr-ft-F	Elastic Modulus $\times 10^{6}$ psi (1000°F)	Combined Stress, psi	Thermal Stress, psi
Reaction Bonded Silicon Nitride	1.2	6.4	23.1	3300	5250
Hot Pressed Silicon Nitride	1.2	10.5	44.0	5120	7070
Sintered Silicon Nitride	1.7	9.7	40.0	6750	8720
Reaction Sintered Silicon Carbide	1.9	38.4	54.0	6490	6580
Sintered Silicon Carbide	2.6	19.3	56.5	11,000	12,990
Magnesia Alumina Silicate	1.1	0.85	17.4	7750	9660
Fused Silica	0.3	1.2	21.0	700	2610

a conventional aluminum base. A stack of Belleville springs is provided to compensate for longitudinal differences in thermal expansion. A unique design feature is a stack of highly insulating disks between the top and the body, which allows the use of high strength ceramics (silicon carbide), which have a relatively high thermal conductivity, and can withstand high temperatures at moderate compression loads. The insulating properties of the disks were achieved by introducing a high degree of surface roughness which creates an efficient thermal barrier when the disks are used in a stack. According to Ref. 51, a hot pressed, silicon nitride capped piston was operated for 164 hours under operating conditions that produced a BMEP of 245 psi, a 20.4 A/F ratio, and a peak pressure of 2500 psi cylinder pressure. The test was discontinued because of bolt loosening problems.

Another approach that essentially circumvents the problems with ceramic poppet valves is that taken by Johnston (Figures 5.8-5 and 5.8-6). The Johnston engine takes advantage of air motion and the potential of the two-cycle concept for the application of ceramics. The Johnston engine is generally of conventional single piston uniflow design but uses a cylindrical "cookie cutter" type of sleeve valve instead of poppet valves to circumvent the stress and impact problems with poppet valves. Swirl is introduced through the ports of the sleeve valve and is enhanced during final compression by the convergence of the space between the piston crown and the cylinder head, which produces vortex acceleration and spiraling squish flow. Figure 5.8-7 compares the projected power versus fuel economy produced with the concept as compared to current technology. The fuel economy gains shown are fairly consistent with those predicted by Cummins (Figure 5.8-3).

5.9 HARDWARE DESIGN

In the past, development of diesel engine hardware was primarily dependent upon testing techniques which concentrated on eliminating weaknesses that would otherwise jeopardize the program objectives. Such development approaches have frequently led to an unbalanced design, with certain components over-designed or containing redundant structures and mechanisms. As shown in Figure 5.9-1, considerable improvement in specific weight has been achieved between 1920 and the Second World War, primarily from the combined results of combustion efficiency improvements, and the introduction of lightweight materials and improvements in foundry technology. Progress leveled off because of limitations of analytical and material resources. The advances in computer technology and the adoption of "Computer-Aided Design" (CAD) techniques are partly responsible for the renewed reduction of engine specific weight after 1960, and it is expected that further refinements in CAD approaches will make additional improvements possible as indicated by the projection in Figure 5.9-1. As can be seen in Figure 5.9-2, the more comprehensive use of CAD will also considerably shorten the learning period that each engine project must go through before it reaches the production stage.

Complex computer programs are now available to the engineer which employ digital and analog simulation techniques, and which will be helpful in resolving dynamic, heat flux, stress and deformation problems in critical

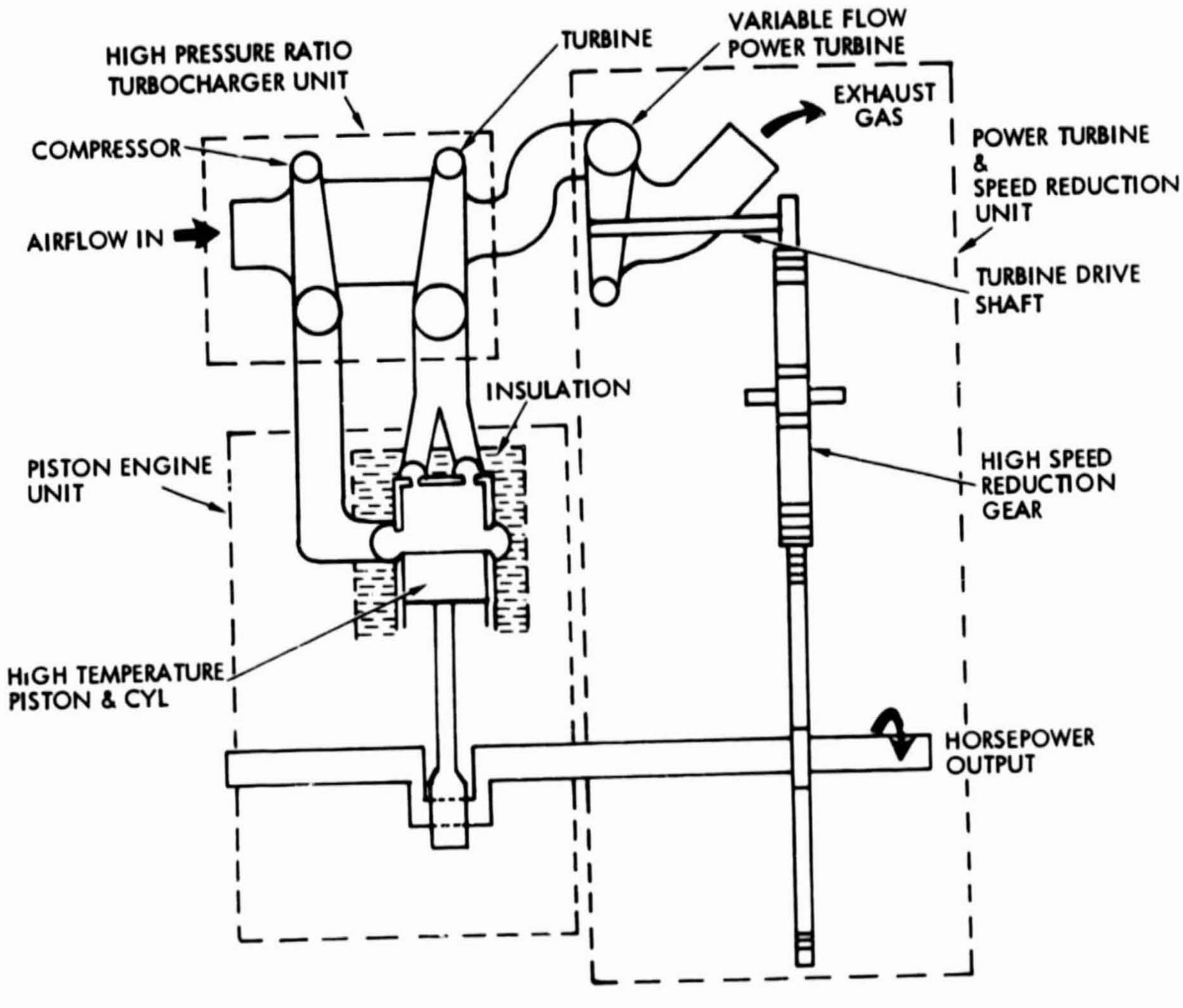


Figure 5.8-5. Johnston Engine Schematic (Ref. 53)

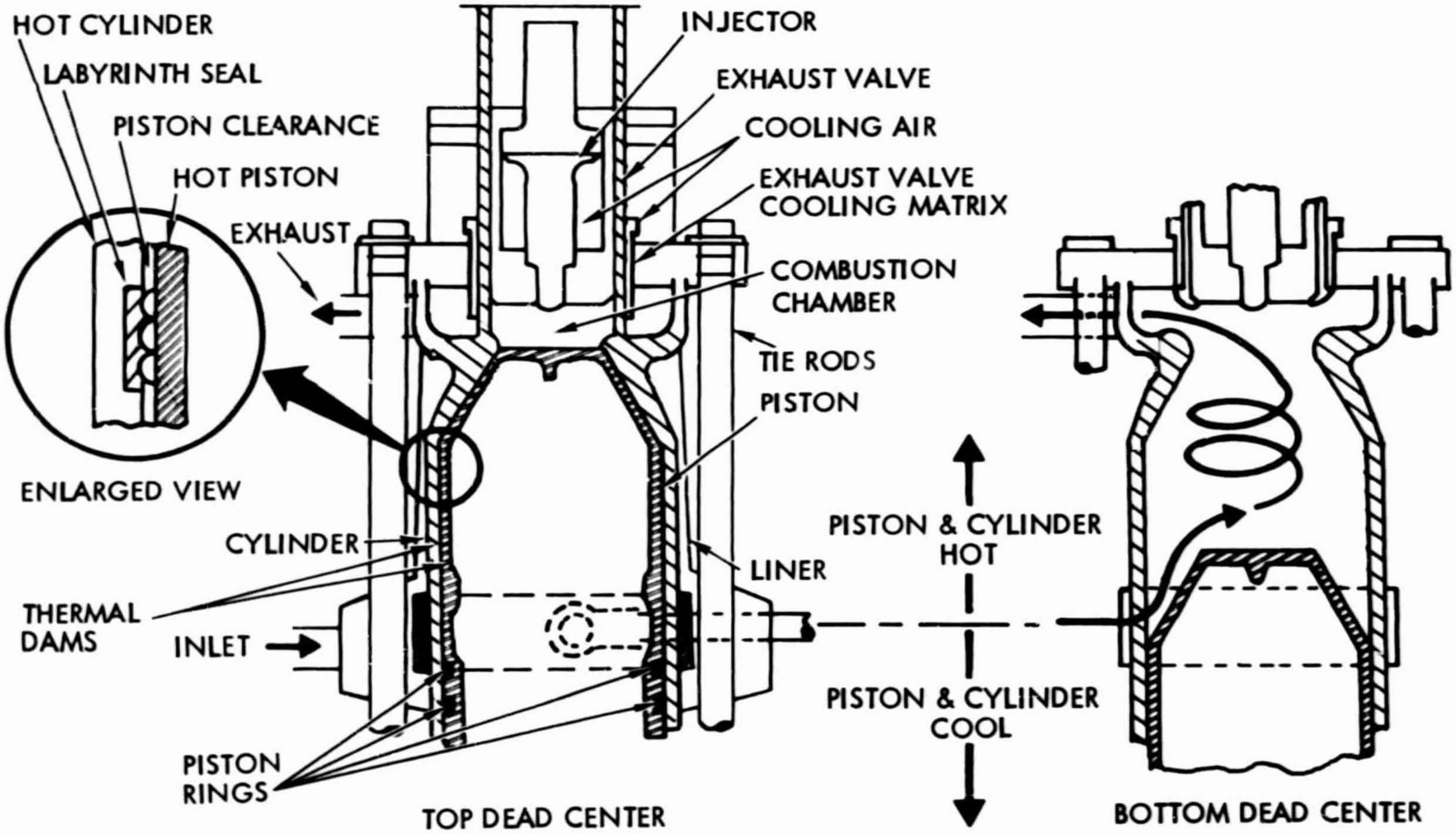


Figure 5.8-6. Johnston Engine Cylinder and Piston Configuration (Ref. 53)

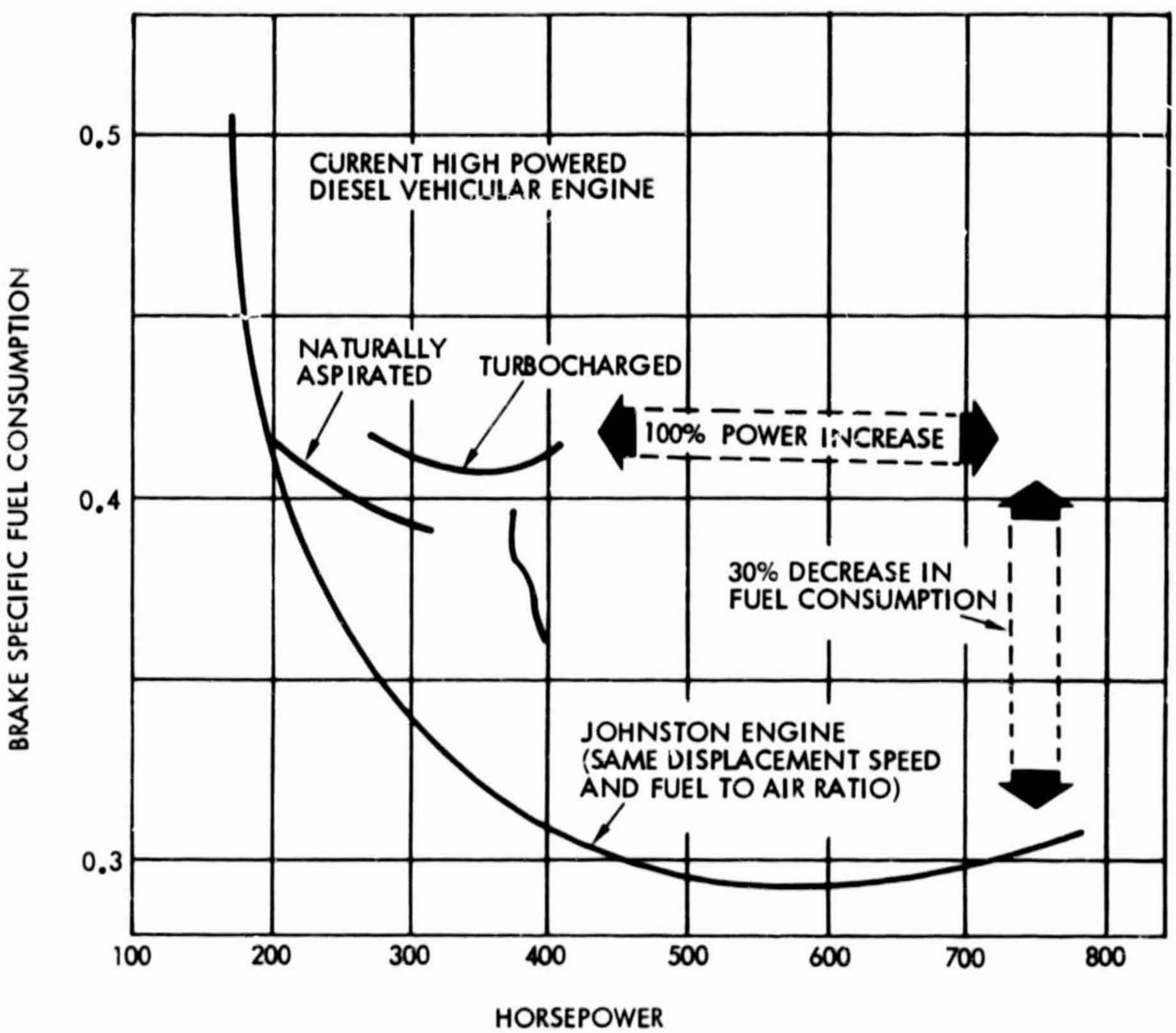


Figure 5.8-7. Comparison of Power/Fuel Economy Relationships (Ref. 53)

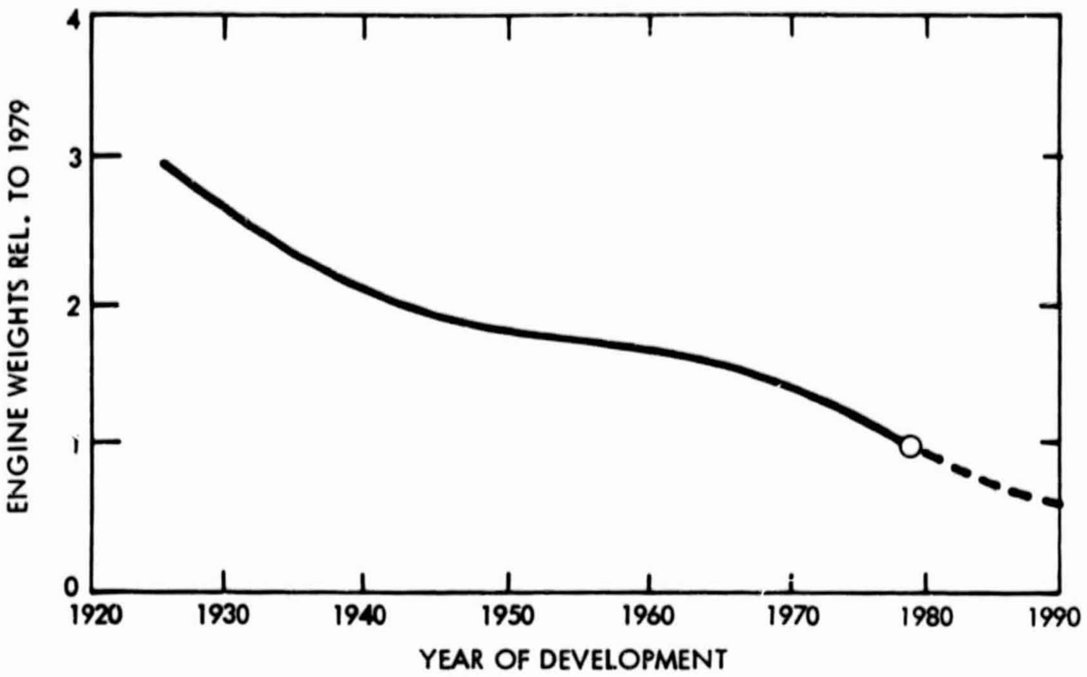


Figure 5.9-1. Diesel Engine Weight Reduction with Time (Ref. 40)

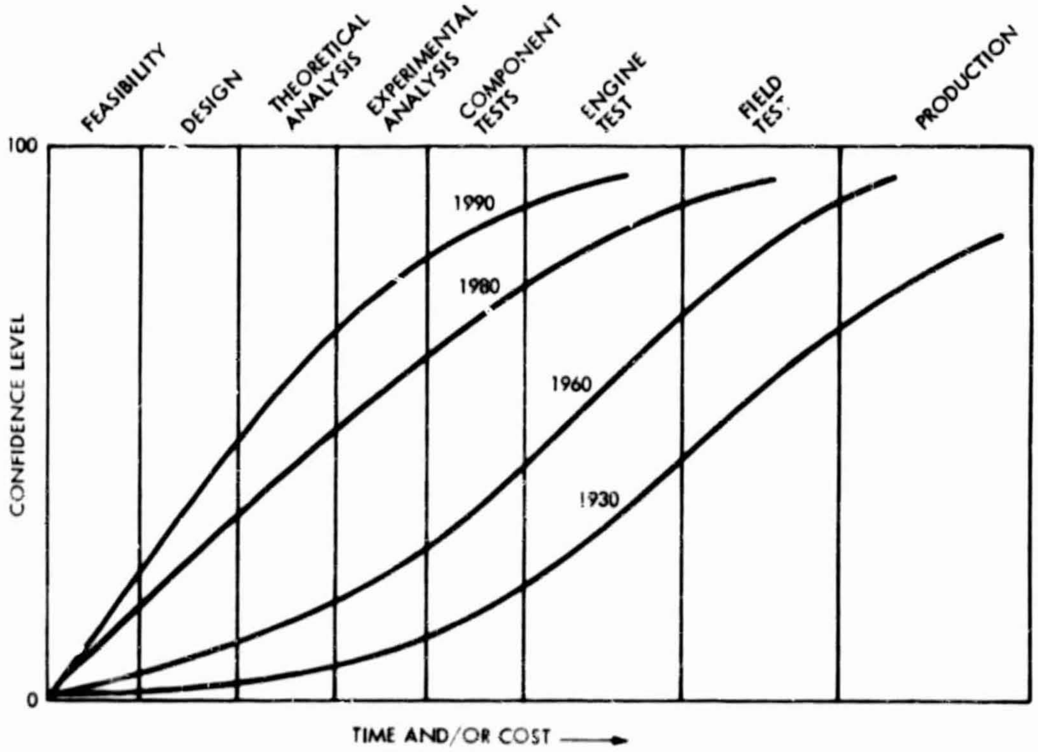


Figure 5.9-2. Change of Learning Period with Time (Ref. 40)

components such as pistons (Figure 5.9-3), push rods (Figure 5.9-4), crank shafts, fuel lines, etc. The use of CAD will also be helpful in pin-pointing, reducing and eliminating the primary sources of noise development caused by housing deflections as shown in Figures 5.9-5 and 5.9-6.

Further reduction in engine weight is also expected to come about with the introduction of advanced light-weight materials and improved production technology. Of course, the discussed hardware improvement potential applies to the gasoline engine as well, but being a more stress critical piece of hardware, the diesel is expected to benefit considerably more from the application of CAD techniques.

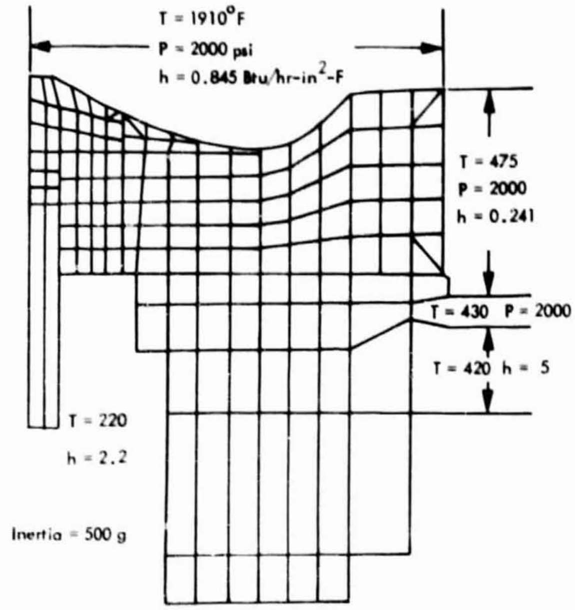
5.10 FRICTION

In terms of total energy, the engine friction as can be seen from Figure 5.1-1 and Table 5.1-1, is not a large contributor to total losses. However, because it is a direct trade-off between work exerted on the piston and applied to the crankshaft, mechanical friction has a strong effect on fuel economy, engine speed rating, specific weight and size. According to competent literature (Ref. 55) diesel engines have a considerable potential for reduction of friction to the extent indicated in Figure 5.10-1. As illustrated in Figure 5.10-2, this potential can be used to either improve power output and fuel consumption up to approximately 10%, or to increase the speed rating of the engine by 25%, with the resultant benefit of reduced size and weight.

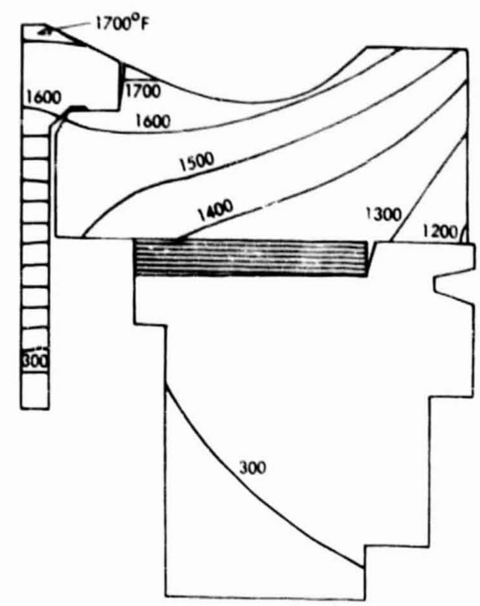
At this time, there is very little known about the physical nature of friction, the total energy actually wasted because of friction under actual operating conditions, or the degree to which various design parameters control frictional losses. Available friction data are obtained from motoring the engine and from extrapolation through the so-called Willansline (Figure 5.10-3). Speed rating has a prominent effect on friction losses. Engine friction also depends to a great extent on oil viscosity which, in turn is strongly affected by oil temperature.

In addition to using a low friction oil, the primary approach to lower friction is through increased and more tightly controlled oil temperature during all operational conditions. Unfortunately, with increasing temperature the minimum film thickness also decreases and, as shown in Figures 5.10-4 and 5.10-5, requires closer tolerances or larger bearing areas. This means that most measures to reduce friction also lead to increased engine dimensions and weight, and cannot be indiscriminately used without making an engine too heavy and bulky. Extensive design and cost trade-off studies are required for optimization. It has been estimated that engine friction can be reduced by as much as 40% with an optimized design, which would improve fuel efficiency by approximately 12% (Ref. 55).

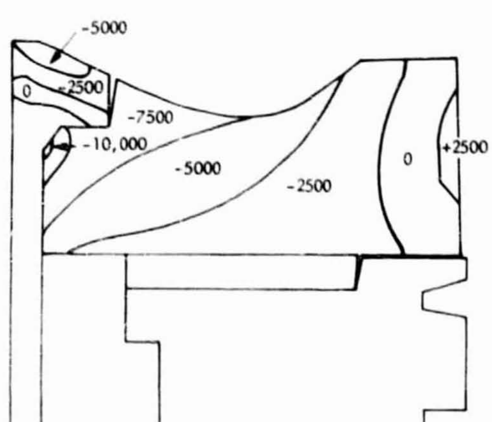
The application of roller and needle bearings is also a means of reducing friction. Roller bearings are used for the crankshaft in a number of large vehicular diesel engines, such as the Maybach engine shown in Figure 5.10-6. Maybach diesel engines have an undivided tunnel type crankshaft housing, and the crankshaft bearing assembly is inserted in one piece from the engine



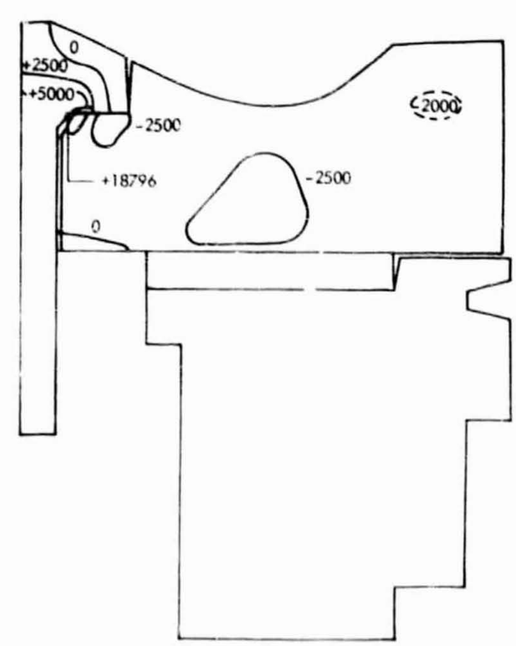
FINITE ELEMENT GRID NETWORK AND BOUNDARY CONDITIONS FOR ADIABATIC ENGINE PISTON



TEMPERATURE DISTRIBUTION IN AN ADIABATIC ENGINE PISTON WITH A REACTION BONDED SILICON NITRIDE TOP

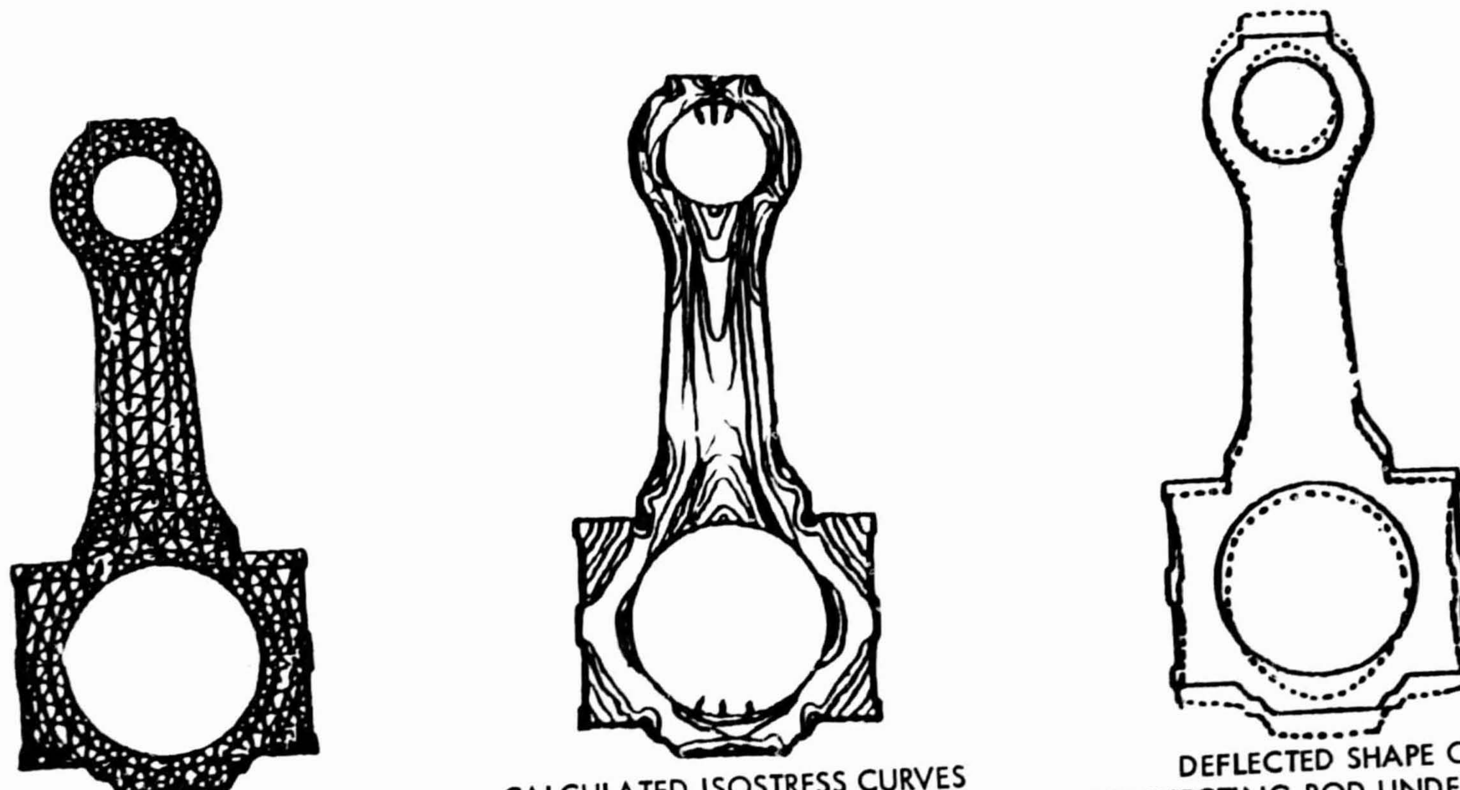


NORMAL STRESS IN AN ADIABATIC ENGINE PISTON WITH A REACTION BONDED SILICON NITRIDE TOP



MAXIMUM PRINCIPLE STRESS IN AN ADIABATIC ENGINE PISTON WITH A REACTION BONDED SILICON NITRIDE TOP

Figure 5.9-3. Application of Advanced Computer Techniques in Stress Analysis of Ceramic Pistons (Ref. 51)



FINITE ELEMENT MODEL OF CONNECTING ROD

CALCULATED ISOSTRESS CURVES FOR CONNECTING ROD

DEFLECTED SHAPE OF CONNECTING ROD UNDER LOAD

Figure 5.9-4. Application of Advanced Computer Techniques in Design of Connecting Rods (Ref. 40)

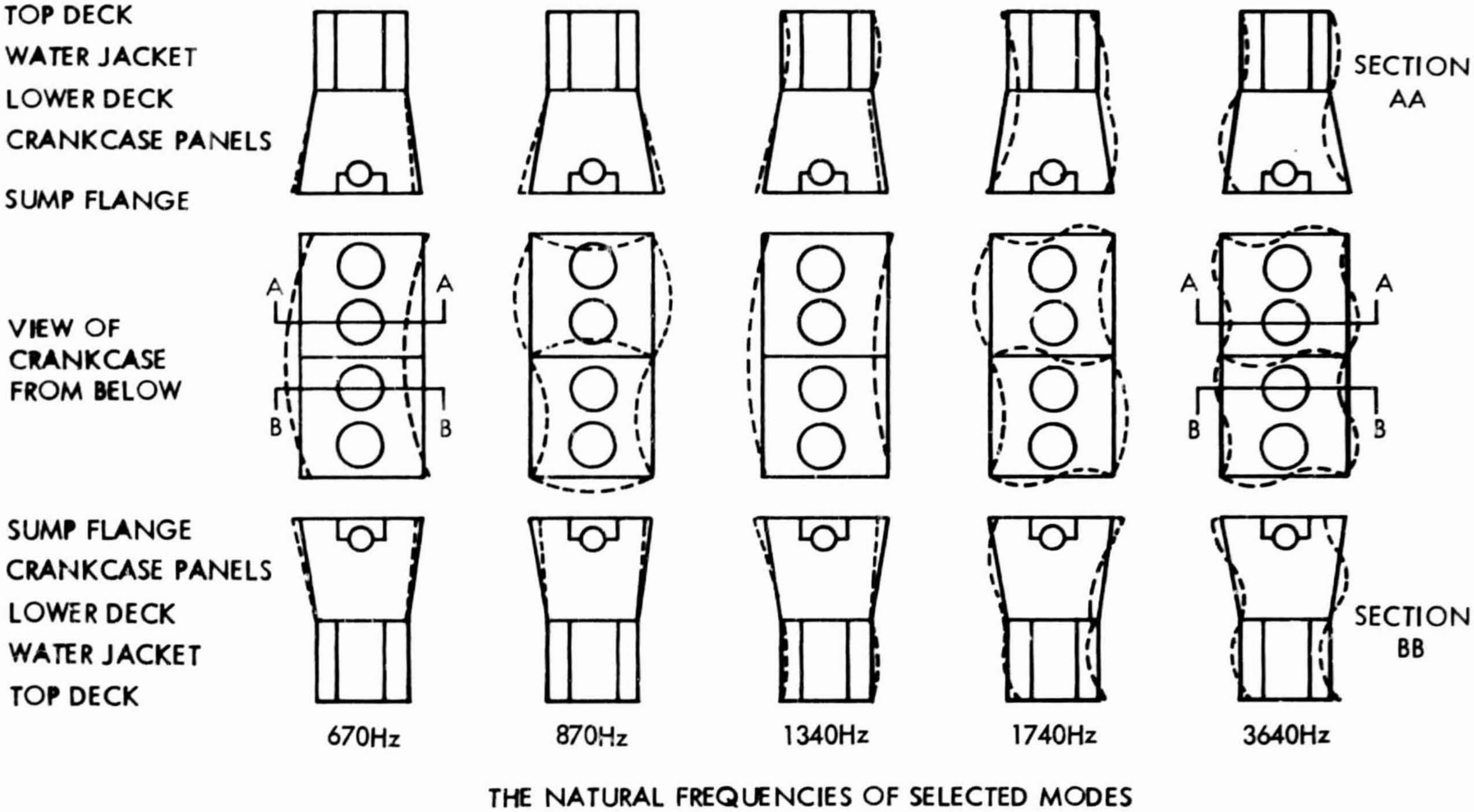


Figure 5.9-5. Typical Modes of Crankcase Deflection Primarily Responsible for Diesel Noise Generation (Ref. 54)

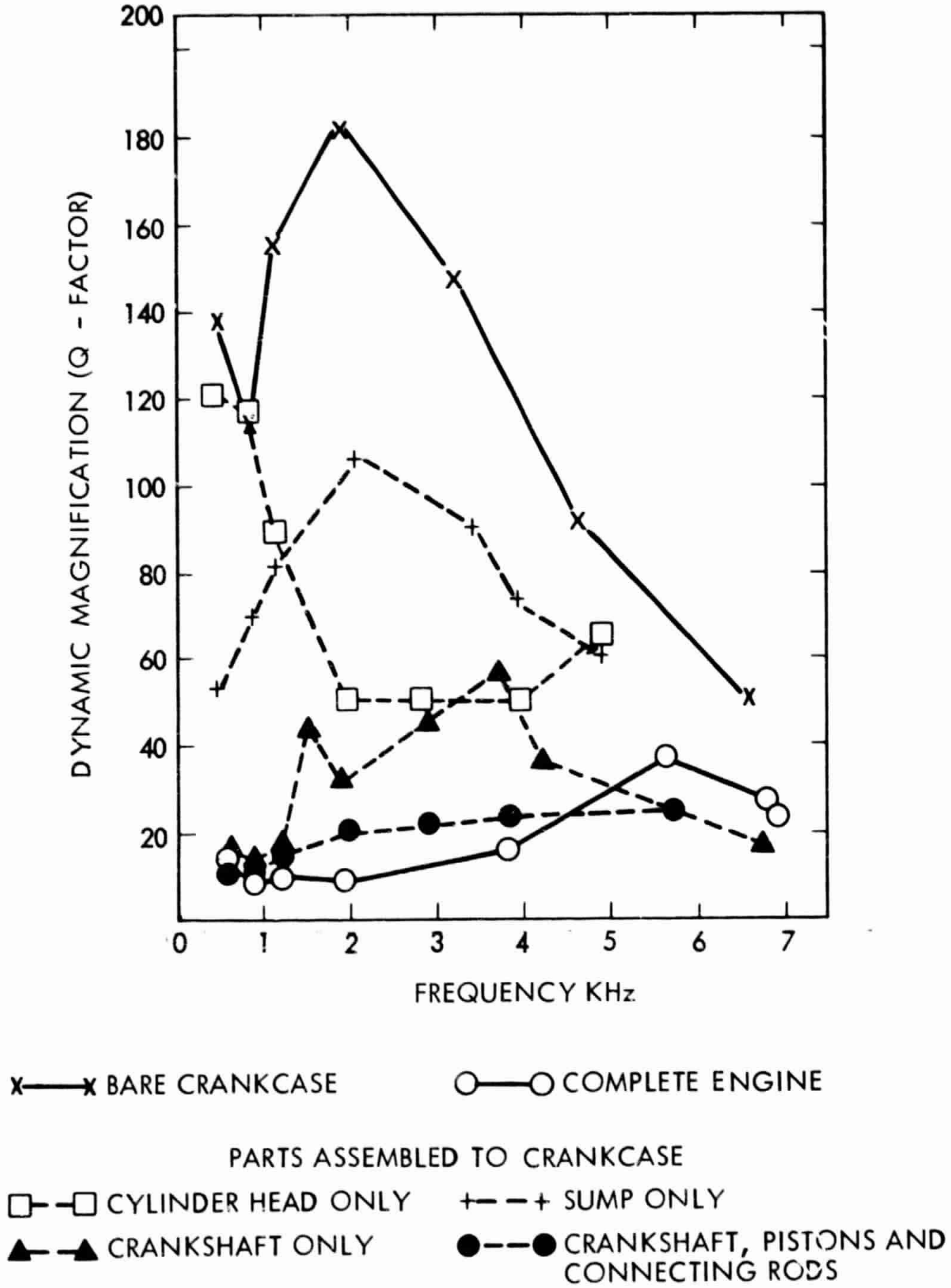


Figure 5.9-6. Effect of Structural Damping on Diesel Engine Noise (Ref. 54)

FRICTION-MEP (TORQUE)

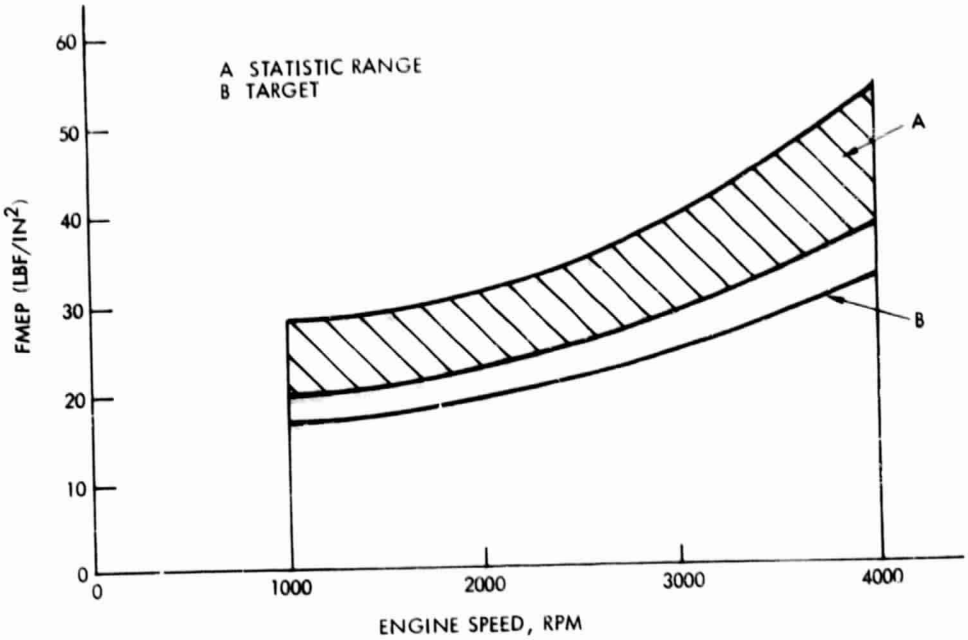


Figure 5.10-1. Friction Mean Effective Pressure Loss vs. Engine Speed (Ref. 55)

REDUCTION OF FRICTION BY 10% PERMITS INCREASE OF RPM AND POWER BY 25% AT SAME BSFC LEVEL

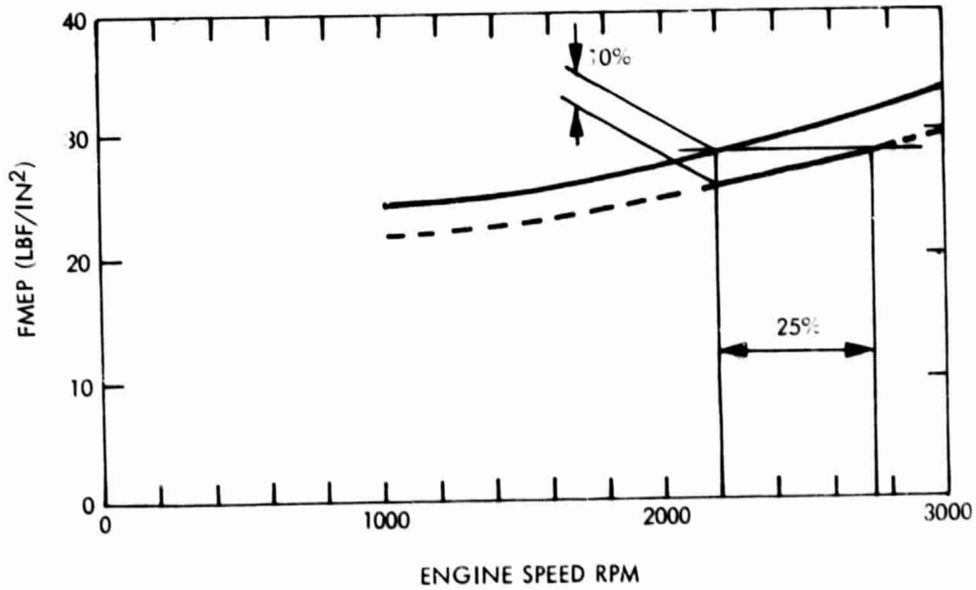


Figure 5.10-2. Benefits of Reduced Friction in Terms of Mean Effective Pressure and/or Engine Speed (Ref. 55)

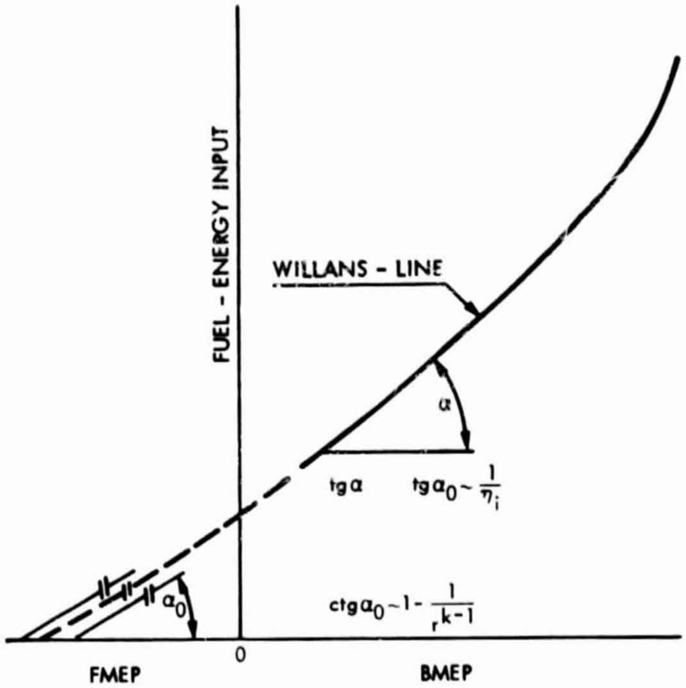


Figure 5.10-3. Determination of Internal Friction Losses from Motoring Characteristics (Ref. 55)

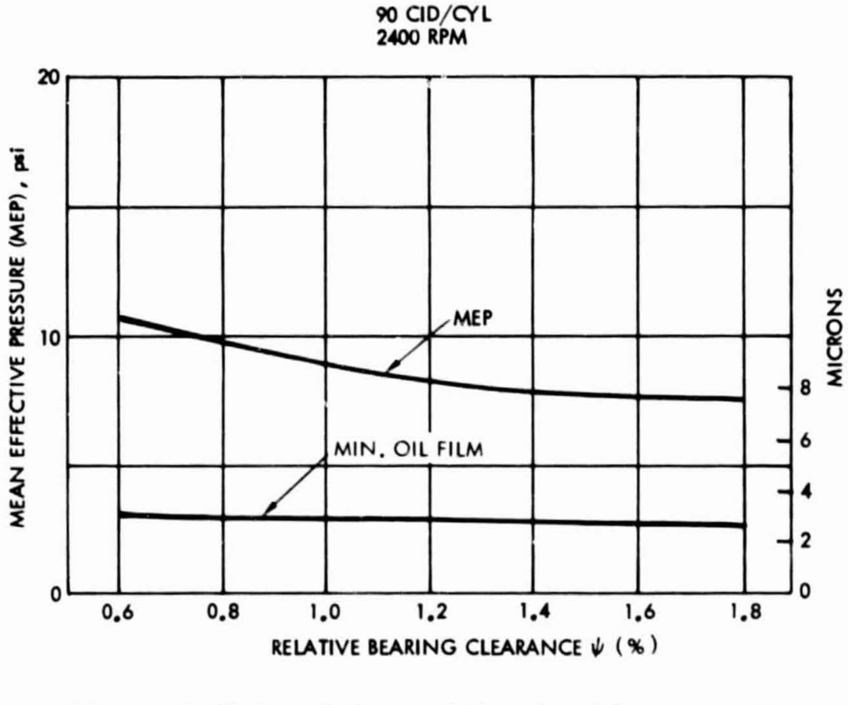


Figure 5.10-4. Effect of Bearing Clearance on Friction Losses (Ref. 55)

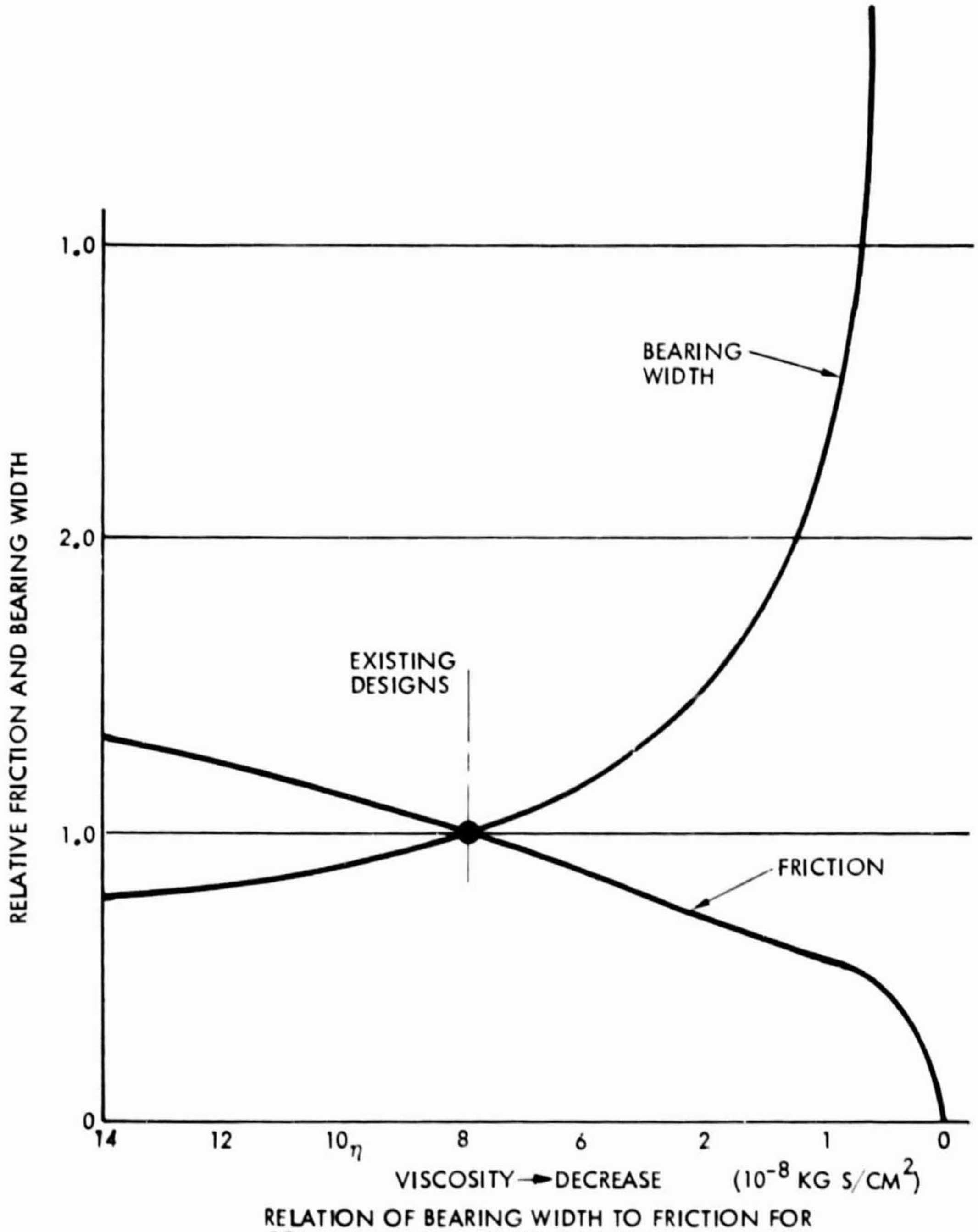


Figure 5.10-5. Effect of Oil Viscosity on Bearing Friction and Width (Ref. 55)

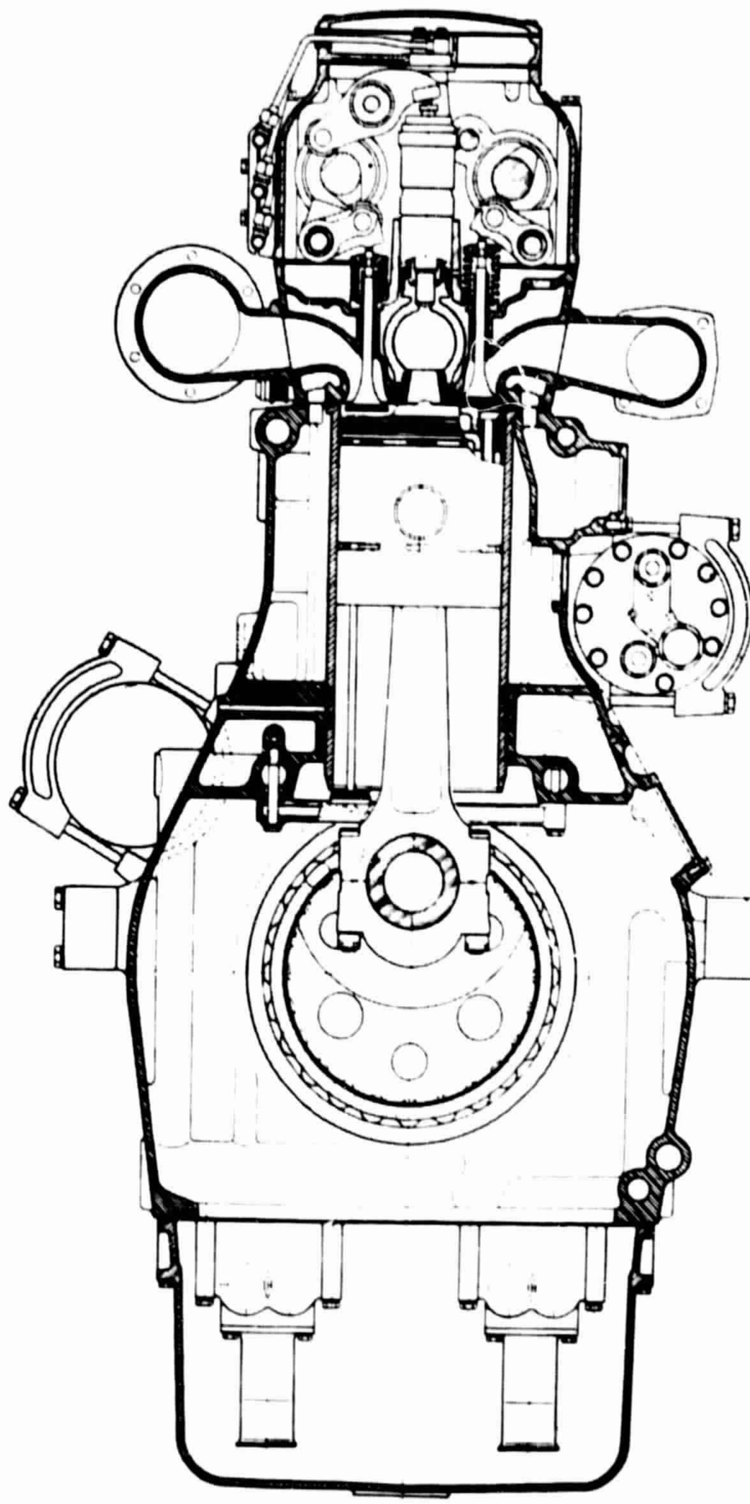


Figure 5.10-6. Maybach Type MD-320 Diesel Engine (Ref. 56)

Note the Large Diameter Roller Journal
Bearings and Undivided Crankcase

end. To accomplish this, the bearings must be fitted over the journals of the crankshaft. Approaches in this direction are usually undesirable for automobile engines from the cost standpoint, but should be a part of friction-oriented design optimization studies. Because most efforts to significantly reduce the friction of conventional bearings lead to increased engine dimensions, there might well be a cross-over point at which the introduction of roller bearings is favorable, from cost standpoint as well.

5.11 THREE-CYLINDER ENGINES

A three-cylinder engine can conceivably be of advantage in the small diesel class of about 1-ℓ to power two-seated subcompact commuter cars having an inertia weight of about 1600 lb, and a fuel economy of 60 + mpg for a natually aspirated, and 80 + mpg for turbocharged production duction engines. According to the Volkswagen design trade-off study, discussed in Section 3.4, the choice of the number of cylinders has a noticeable effect on engine specific weight and power concentration (Figures 3.4-8 and 3.4-9). This relationship becomes increasingly pronounced with single cylinder displacement sensitivity to power concentration brought about by the composite effect of size-related internal aerodynamics and thermal/mechanical limitations, such as piston speed, boreto-stroke ratio, etc. In the case of VW, the single cylinder displacement chosen for their family of engines was 400 cc, which led to the need for five cylinders for the 2-ℓ class. Extrapolating the results of this study of small displacement engines downward from the 1.6-ℓ Rabbit engine, the same study suggests that three-cylinder engines could conceivably be of advantage in the 1-ℓ displacement class.

Reasoning in this direction is not new. Loop-scavenged, two-cycle spark-ignition engines, for example did show a strong sensitivity to cylinder displacement with a distinct optimum at about 300 cc per cylinder. Three cylinders were therefore chosen for the Auto Union two-cycle engine that powered the German Auto Union (DKW) and Swedish Saab passenger cars for more than two decades (1938 to 1963). According to recent press information VW, Ford, Buick, and General Motors have voiced plans to introduce three cylinder gasoline and diesel engines for computer mini cars between 1985 and 1990.

5.12 TWO-CYCLE ENGINES

Two cycle diesel engines are widely in use for marine applications, railroad locomotives and trucks. Their performance and fuel economy is competitive with their four cycle counterparts although they are considerably lighter. All of the two-cycle diesels in use are of direct injection and open chamber design and are uniflow scavanged, using a combination of piston controlled inlet ports and poppet outlet valves, or piston only controlled inlet and outlet ports. Mechanically driven Roots or centrifugal blowers are commonly used for scavenging on two-cycle vehicular diesels, with or without the additional use of tubochargers. The operating speed of all two-cycle diesels in use is generally low, below 2500 rpm.

One of the major reasons why high-speed, two-cycle automotive diesels are not in use today is that two-cycle engines are more difficult to develop. An approach to the high-speed automotive diesel through the two-cycle concept would have meant an unnecessary compounding of problems which developers were reluctant to undertake. This does not mean that two-cycle engines have no potential for light and medium duty vehicular application in the future. Two-cycle engines of the valve or opposed piston uniflow type definitely deserve renewed attention and re-evaluation, taking current and projected technology advances into account.

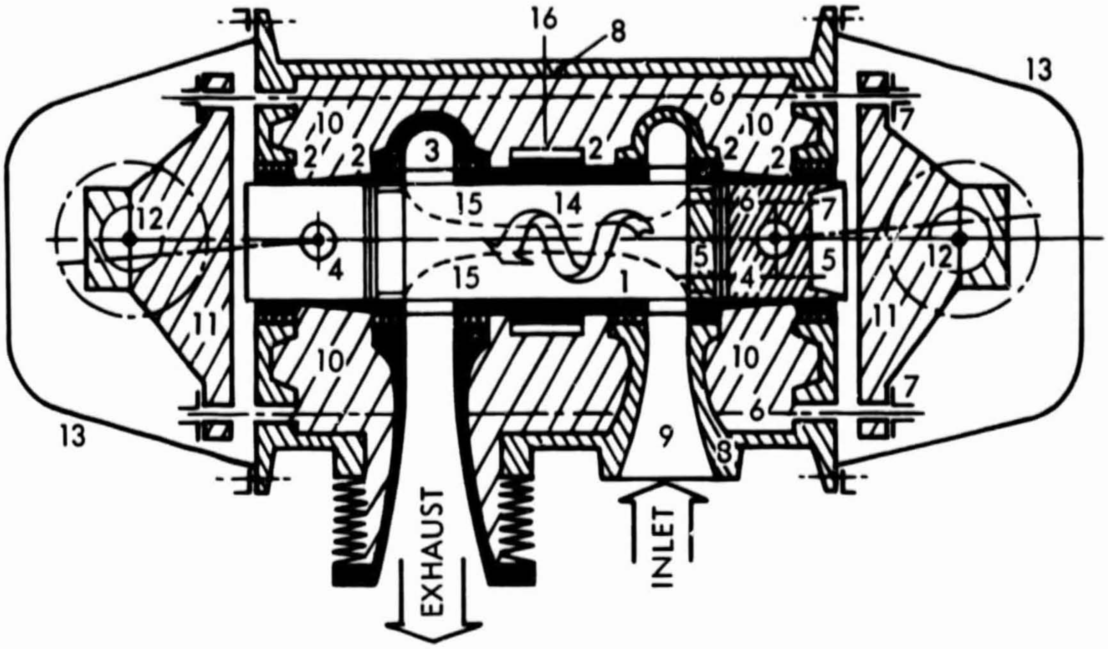
A well known problem with two-cycle engines is that of residual gases. This is primarily responsible for the relatively poor volumetric efficiency of two-cycle engines as compared to four-cycle engines. With EGR possibly becoming a necessity in diesel engines to meet future emission requirements, the residual gas problem of the two-cycle engine can possibly be used to its advantage. The two-cycle engine already has a built-in EGR capability. The amount of residual gas entrapment can be varied over a relatively wide range by controlling the inlet pressure and back pressure of the engine. No additional systems that can contaminate and fail are required to implement and to control EGR.

Because of their simplicity, two-cycle engines, in particular those of the opposed piston uniflow type, lend themselves better to the application of ceramic materials than the more complex configuration of four-cycle cylinder heads. The latter require the accomodation of at least two poppet valves. Small two-cycle engines will require only one poppet valve in the center of the head. In opposed piston type engines, there is no cylinder head at all. All interior surfaces are cylindrical and essentially flat.

The opposed piston two-cycle engine concept has a number of other advantages. As shown schematically in Figure 5.12-1, the major loads between pistons and crankshafts can be balanced by using tie-rods without imparting strain, vibration and shock loads to the housings. The sleeve can be free-floating, and compressive hoop stresses may possibly be introduced by wrapping the ceramic sleeve with glass fiber.

Aerodynamically clean radial swirl can be induced with the inlet ports, which is necessary to achieve low NO_x emission in open chamber stratified combustion. Experience with uniflow opposed piston engines has shown that without the use of swish techniques the residual swirl intensity is sufficient to produce a half-turn of the compressed air volume during injection and to generate a radial gravity field that rapidly displaces hot combustion products toward the center of rotation with a minimum of turbulence and mixing. This should be favorable from the NO_x stand point.

The need for a synchronizing gear train between the two crankshafts has been a major deterrent in the adoption of opposed piston concepts by developers of commercial diesel engines. With recent advances for small engines in the design of cogged nylon belts, this problem may possibly be overcome in a cost-effective and economically viable way. Attempts to circumvent the two crank synchronization problem have been made by introducing additional rocker arms and push rods that connect the pistons to a single common crankshaft (Figures 5.12-2 and 5.12-3). This results in a relatively bulky



LEGEND:

1. ONE PC. CERAMIC CYLINDER SLEEVE, COMPOSITE, HOT PRESSED
2. FILAMENT WOUND HOOPS
3. CERAMIC EXHAUST PLENUM
4. CERAMIC PISTONS
5. PRESSURE PLATES
6. TIERODS
7. TEMPERATURE COMPENSATING SPACERS
8. METALLIC HOUSINGS
9. INLET PLENUM
10. INSULATED SPACE
11. YOKES
12. CRANKSHAFT BEARINGS
13. LIGHT WEIGHT COVERS
14. HELICAL SCAVENGE FLOW
15. RESIDUAL HOT GAS CORE

Figure 5.12-1. Conceivable Concept of an Adiabatic Opposed-Piston Two-Cycle Engine

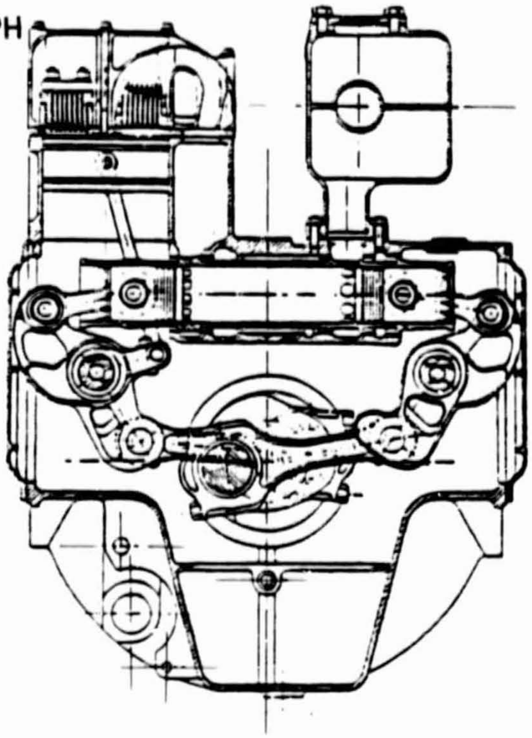


Figure 5.12-2. Naturally Aspirated Single Crankshaft Two-Cycle Uniflow Diesel Engine

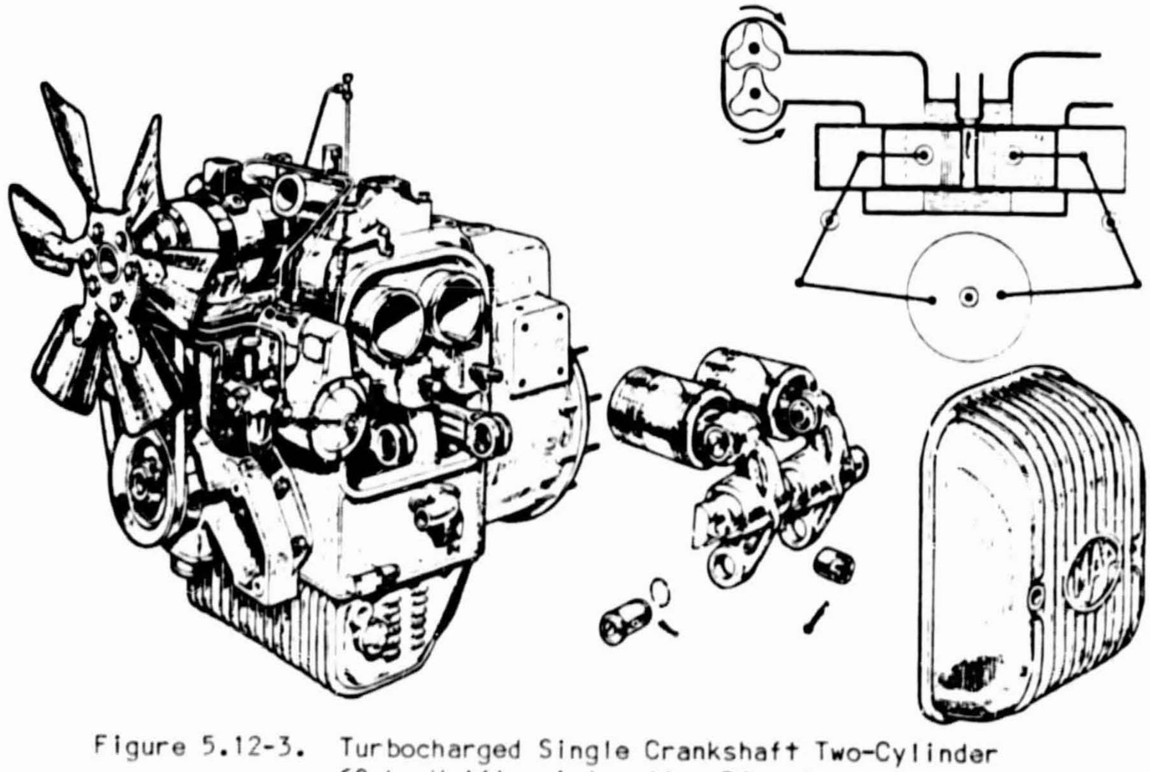


Figure 5.12-3. Turbocharged Single Crankshaft Two-Cylinder 60 hp Uniflow Automotive Diesel, Manufacture d'Armes de Paris (M.A.P.) (Ref. 57)

engine. But as described in Section 5.7 (Figure 5.7-3), it also offers a feasible means of implementing variable compression.

The extensive development background of two-cycle diesel engines which exists, can be usefully applied in assessing their developmnet potential for light and medium duty vehicular uses. In the United States, General Motors and Detroit Diesel Allison have been the leading developers of two-cycle heavy duty Vehicular diesels which are all of the uniflow, multiple outlet valve type as shown in Figure 5.12-4.

The Thiokol Chemical Corp. attempted to develop a 101 CID light duty two-cycle diesel engine for trucks and boats in the early sixties. As shown in Figure 5.12-5 the engine was of a unique radial twin crankshaft design using a U-shaped combustion chamber shared by two adjacent cylinders. The purpose of the dual crankshaft arrangement was to affect piston phasing for improved scavenging and volumetric efficiency. As shown in Figure 5.12-6, the concept would have been very advantageous from the packaging standpoint, but it suffered from a number of inherent drawbacks. These were primarily: (1) an intolerably high heat rejection rate because of a large surface to volume ratio of the combustion chamber, (2) the inability to start on diesel fuel because of cross-over volume between adjacent cylinders limiting the compression ratio to approximately 18:1 and, (3) poor scavenging efficiency because of insufficient piston phasing. The development was therefore discontinued. Ceramic lining of the cylinder heads, which is believed feasible in the near future as well as turbocharging and preheating of the inlet air could possibly have solved the heat rejection and the starting problem which were one of the primary reasons for discontinuing this development work.

Encouraging development work with a four-cylinder, two-cycle radial engine for light aircraft which had been started during the 1950s, was reported by the McCulloch Corp., Los Angeles, in 1970. As shown in Figure 5.12-7, the engine has a unique toroidal combustion chamber which, as McCulloch claims claims, has shown superior combustion efficiency and emission characteristics. Table 5.12-1 summarizes projected performance data derived from tests. Reportedly, evaluation tests with a prototype engine will be conducted by the NASA-Lewis Research Center in Cleveland in the near future. The radial engine concept does not seem to be favorable for vehicle applications; however, the toroidal combustion chamber concept definitely deserves attention.

Opposed piston two-cycle engines of the type shown in Figure 5.12-3 were developed in the 1950s by Sulzer in Switzerland and MAP (Manufacture D'Armes de Paris) in France. Opposed piston diesels for industrial uses of free-piston and dual crankshaft design were developed by Junkers and Pescara in Germany and in France prior to The Second World War. Turbocharged and turbocompounded opposed piston two-cycle diesels for aircraft uses were developed by Junkers in Germany, and by Napier in Britain during and after the Second World War.

Only two isolated research approaches using the two-cycle concept have recently been taken. One approach, aimed primarily at the implementation of variable compression ratio, uses an opposed piston engine design as shown in Figure 5.7-3 (Ref. 50). The approach does not take advantage of the

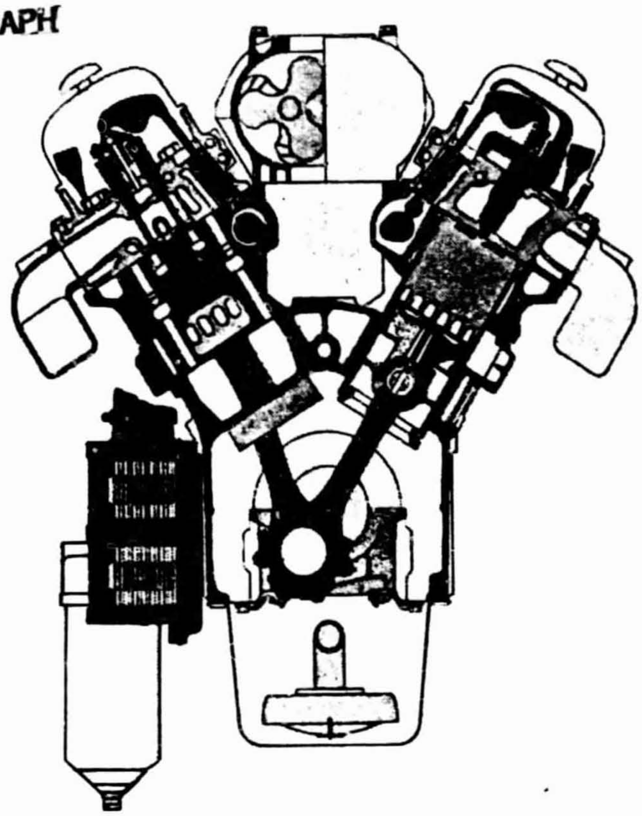


Figure 5.12-4. Cross-Section of Detroit Diesel Allison V92 Two-Cycle Diesel Engine, 92 CID Per Cylinder (Ref. 58)

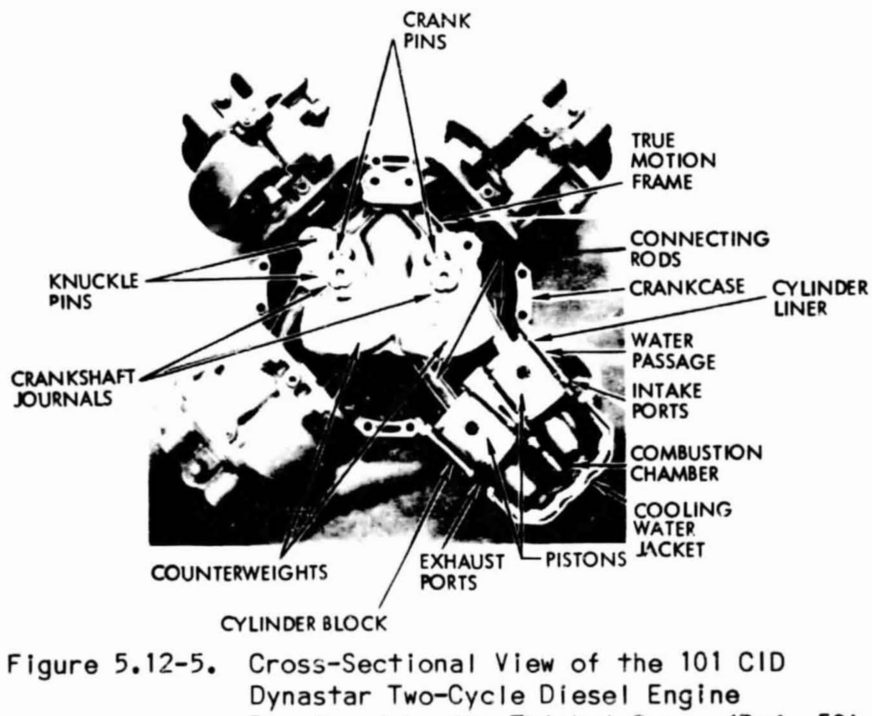


Figure 5.12-5. Cross-Sectional View of the 101 CID Dynastar Two-Cycle Diesel Engine Developed by the Thiokol Corp. (Ref. 59)

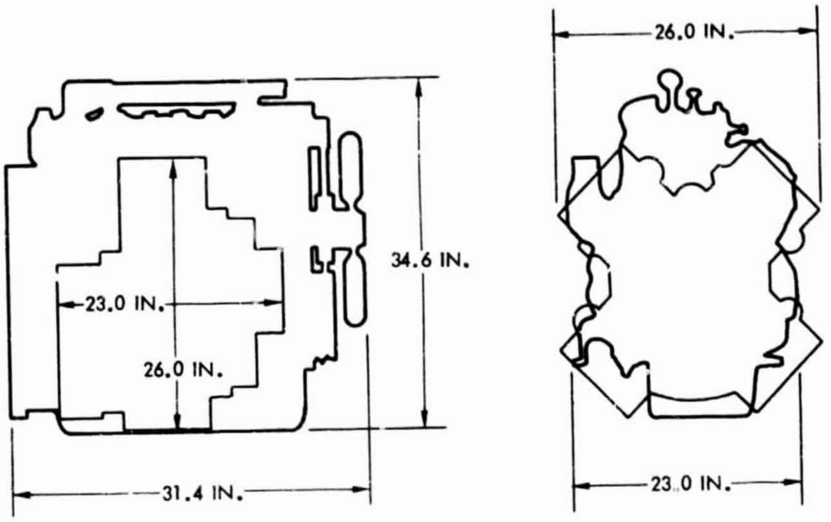


Figure 5.12-6. Size Comparison of Dynastar Radial Diesel Engine with Typical Conventional Diesel Engine of Same Displacement (Ref. 60)

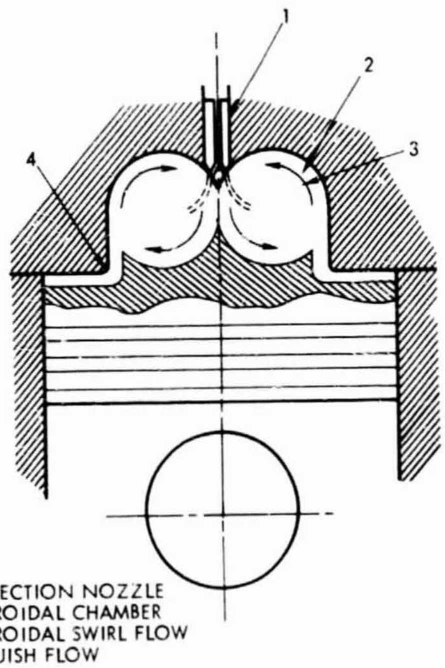


Figure 5.12-7. Schematic of McCulloch Two-Cycle Diesel Engine with Toroidal Combustion Chamber (Ref. 60)

Table 5.12-1. Dynastar Radial Diesel Engine Projected Performance Based Upon Single Cylinder Tests (Ref. 59)

	Dynastar Model No.		
Characteristics	CL430	CL830	CL843
Brake Horsepower	50	100	200
RPM	2500	2500	2200
No. of Cylinders	4	8	8
Bore (in.)	3.0	3.0	4.25
Stroke (in.)	3.5	3.5	4.25
Piston Area (in.^{2})	28.3	56.5	113
Displacement (in.^{3})	99	198	480
Compression Ratio	17:1	17:1	17:1
Weight (lb)	200	400	800
Length (in.)	21	23	27
Height (in.)	17	26	32
Width (in.)	26	26	32
Installation Evelope (ft^3)	5.5	9	16
Lb/HP	4.0	4.0	4.0
HP/ft^3	9.1	11.1	12.5
HP/in^2 Piston Area	1.77	1.7	1.77

concept of stratified combustion and the application of ceramic materials. The approach taken by Johnston is primarily aimed at the implementation of ceramics (Figure 5.8-5). It uses conventional single piston uniflow design, but has a sleeve valve to control the outlet flow, instead of the conventional poppet valve. As described in Sections 5.7 and 5.8, both of the above approaches have shown encouraging results.

In summary it appears that the two-cycle engine concept has a potential for application in high speed automotive diesel engines and definitely deserves attention. In particular, this is true for small displacement three cylinder engines that will display six cylinder four-cycle engine characteristics.

SECTION 6

SUMMARY AND CONCLUDING REMARKS

The major findings made in this study are summarized as follows:

6.1 FUEL ECONOMY

In their present form, diesel-powered cars offer fuel economy advantages to the individual consumer and to the community that make further dieselization of the U.S. fleet of light to medium duty vehicles desirable. Based upon EPA ratings, diesel cars currently in use are 25 to 60% more fuel efficient than currently designed gasoline-powered counterparts. On the basis of sales, the average diesel fuel economy advantage over gasoline-powered cars is 38 percent, but only 25% if compared against gasoline-powered economy cars of the same weight class. The total energy savings of petroleum fuel that will result from dieselization are estimated to be on the order of 4 to 6% with 25% market penetration.

6.2 DRIVEABILITY

The typical characteristic drawbacks of the diesel in regard to noise, smoke, and odor, have been reduced to a degree that is not overly objectionable to the consumer when taking the fuel economy advantages into account. The diesel's inherent sluggishness can now be overcome by turbocharging, at acceptable cost to the consumer because of recent improvements in small turbine production technology.

6.3 SERVICE LIFE

Conventionally designed automotive diesels such as the ones marketed by Mercedes Benz, Peugeot, Perkins, and Nissan, for example, have established a superior record of reliability and lower operating cost. The producers of gasoline - diesel conversions do not make any claims that their diesels will outlast their gasoline counterparts. There is not enough general road experience available at this point in time to make any statements in this regard. The initial mechanical problems encountered with the General Motors/Olds diesel are not necessarily typical for gasoline-diesel conversions.

6.4 MARKET TRENDS

The factors suggesting significant growth of the diesel market in the United States are: (1) for the consumer - an improved fuel economy and the desire to continue driving a larger, safer and more comfortable car, (2) for industry - the need to meet CAFE regulations and still sell larger cars, and (3) for the community - the increase in miles driven per barrel of crude

oil, and the potential of meeting future emission requirements without the use of critical materials, which are now needed in catalytic converters.

Despite the fact that diesel cars are about 10% more expensive and the price of diesel fuel has risen to 87 percent that of gasoline, the demand for diesel cars is growing. Many of the major producers of gasoline-powered cars not marketing diesels now will outfit selected lines of their cars with foreign-made small diesel for the market of the 1980s. Most auto manufacturers also have their own diesel development underway.

A projection is that 25% of all cars marketed in the United States will be diesels by 1985. However, current consumer interest will possibly diminish with progressing dieselization. This may occur because of mandatory refinery changes and increased processing cost that will have to be passed on to diesel users. Regulated diesel fuel prices to benefit overall petroleum energy savings would require subsidies from other fuels.

6.5 EMISSIONS

The primary factor inhibiting and possibly curtailing dieselization of the U.S. fleet is that of emission. Diesel cars have generally met past standards but are now requiring waivers to meet current standards for gaseous emissions. Diesel cars will not comply with the EPA proposed future schedules for the controlling of particulates, NO_x, CO, and HC. Of particular concern is the emission of particulates which are believed to be carcinogenic in nature, though this claim has not yet been proven. Unfortunately the measures known to reduce specific emission constituents are not necessarily compatible with each other. There is unanimous agreement in the diesel community that more fundamental research is required to significantly reduce diesel emissions. There is serious doubt that the goal of the EPA schedule, which is to reduce the problem emissions of NO_x to 1.0 g/mi, and particulates to 0.2 g/mi, can ever be achieved, unless a technological breakthrough is made. Otherwise, a further downsizing of diesel cars below an inertia weight of 2000 lb (less than the VW Rabbit) would be the only solution to the emission problem.

To give the automotive industry more time to clean up their diesel engines, the EPA has granted a waiver that relaxes the 1981 standards for NO_x from 1.0 to 1.5 g/mi for the period of 2 years through 1983. This is 2 years less than the diesel industry had originally asked for.

EPA-imposed emission standards as well as contending gasoline engine developments make the future of the diesel car in the United States very uncertain. European countries are equally concerned about diesel emission. However, this may not inhibit further dieselization of their fleets until alternative automotive fuel sources (liquid fuel derived from coal, oil shale, or biomass, for example) or significantly more efficient alternative powerplants have been developed.

6.6 CONTENDING ENGINE CONCEPTS

In addition to the emission problem, the diesel is also confronted with the problem of keeping the fuel economy lead over its gasoline fueled contenders which are also rapidly improving. Challenging the diesel in particular in the gasoline engine field are cylinder cut-off techniques, programmed fuel injection, turbo and supercharging techniques, and improved combustion chamber designs with lean burn that will allow for the integration of compression ratios as high as 15:1.

6.7 POTENTIAL FOR IMPROVEMENT

The diesel still has significant potential for fuel economy improvement (40 percent) over the present state of the art. To utilize it, however, will require a great deal of further research. Primary areas with the largest improvement potential are those of open chamber combustion, engine insulation, turbocompounding, and hardware design optimization for the further reduction of structural weight and friction losses.

Advanced diesel R&D work in progress is primarily aimed at open chamber combustion and the use of ceramic materials to achieve hotter walls; which reduces cooling losses, HC and particulate emissions, and smoke development. The initial effort of advanced diesel work will probably be on truck diesels first and become appilcable later to smaller displacement diesels when economically feasible.

Two-cycle engines deserve renewed attention in advanced diesel R&D work because of their potential for high EGR capability, open chamber combustion, and their simple configuration which makes the application of ceramic materials more feasible. Weight and packaging advantages particularly for the valveless opposed piston type could make the two-cycle diesel attractive.

A definite need exists for more fundamental rather than product-oriented research. Resolution of the particulate emission problem especially requires relatively high initial investments in special equipment for high speed combustion analysis, flow visualization and for instantaneous particulate sampling. Broadly based, well coordinated research programs with long-term funding are, therefore, believed necessary to solve the diesel emission problem, and to fully use the known fuel economy potential of the diesel engine.

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